Demand controlled ventilation for multi-family dwellings

Demand specification and system design

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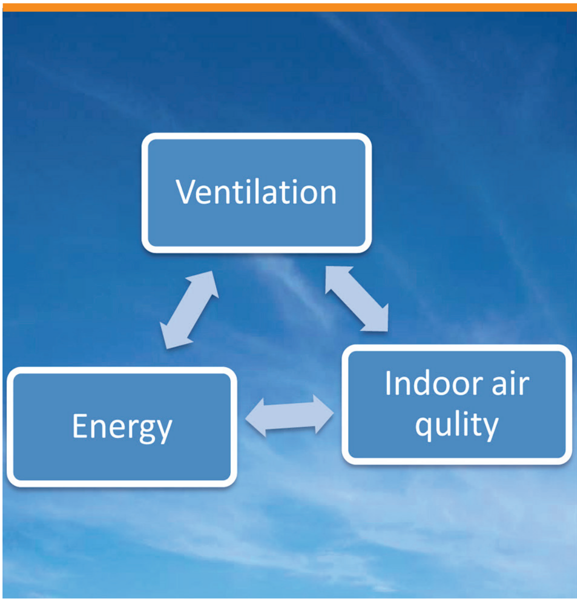

Dorthe Kragsig Mortensen

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Preface

This thesis is submitted as a partial fulfilment of the requirements for the Danish Ph.D. degree. The work has been carried out at the Section of Building Physics and Services at the Department of Civil Engineering at the Technical University of Denmark (DTU). The project was financed by a scholarship from DTU and with supplementary funding from Exhausto A/S. Associate Professor Toke Rammer Nielsen has been the main supervisor of the project, while Professor Svend Svendsen and Adjunct Professor Lars D. Christoffersen have been co-supervisors. In 2009 Max H. Sherman hosted a 6 months external research stay in the Energy Performance of Buildings group at Lawrence Berkeley National Laboratory (LBNL) in connection with the study. The work is based on scientific papers that are summarized in the first part of the thesis. Papers are appended in the second part of the thesis.

Lyngby, April 2011

Dorthe Kragsig Mortensen

Dorthe Kragsig Mortensen

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I would like to thank my team of supervisors for support during the process of writing this thesis and in particular my main supervisor Toke R. Nielsen for sharing knowledge, ideas and time. The gratitude is also extended to DTU and Exhausto for their financial support. I am grateful for my stay at LBNL where I learned many useful things about ventilation and research. Special thanks are given to Max Sherman and especially Iain S. Walker. Thanks to my colleagues at the section of Building Physics and Service and at the International Centre for Indoor Environment and Energy for friendly discussions and good company.

Abstract

The present thesis "*Demand controlled ventilation for multi-family dwellings*" constitutes the summary of a three year project period during which demand specification and system design of demand controlled ventilation for residential buildings were studied.

Most standards and buildings codes specify desired levels of indoor air quality through ventilation rate requirements. The Danish Building Code requires a constant air flow rate equivalent to at least 0.5 air changes per hour in residential buildings. A constant air flow requirement is inconsistent with the time varying needs for ventilation in residential buildings that depend on occupancy, pollutant emission, etc., and results in periods with poor air quality and/or unnecessary energy consumption. If the ventilation rate is varied according to the demand, the indoor climate can be improved and the energy consumption for ventilation can be reduced compared to a system with constant air flow.

A literature study on indoor pollutants in homes, their sources and their impact on humans formed the basis for the demand specification. Emission of pollutants in residential buildings roughly fall into constantly emitted background sources and step-wise constantly emitted sources related to occupancy and activities. Theoretical analyses of these two sources showed the air quality implications associated with the time-varying air flow rates in an occupancy based demand controlled ventilation (DCV) system in comparison to the required constant air flow rate. These analyses were also used to describe the potential air flow savings associated with occupancy based DCV that provide the same average occupant exposure as a system with constant air flow. Results showed that air flow saving up to 26% can be achieved in occupancy based DCV systems compared to systems with constant air flow rates. The trade-off is an increase in peak concentration. However, the time-varying air flow rates of the DCV system are not expected to introduce problematic acute conditions. The issue of system design was focused on simple and cost-effective solutions for centrally balanced DCV systems with heat recovery. A design expected to fulfill this requirement was investigated in detail with regard to its electricity consumption by evaluating a control strategy that resets the static pressure set point at part load. The results showed that this control strategy can reduce the electricity consumption by 20% to 30% compared to a system with fixed static pressure control.

The results of the project provide more flexible approaches to ventilation design for residences that allow occupancy based DCV approaches to comply with codes and standards that are currently based on continuous ventilation rates. Furthermore, a simple, cost-effective and energy-efficient system design for DCV in multi-family dwellings is proposed.

Resumé

Nærværende opsummering af afhandlingen "*Behovsstyret ventilation til etageboliger*" afslutter en treårig projektperiode, hvor behovsspecifikationer og design af systemer til behovsstyret ventilation i boliger er blevet undersøgt.

De fleste standarder og bygningsreglementer specificerer ønsket luftkvalitet igennem krav til tilførslen af udeluft. Det danske bygningsreglement kræver en konstant udelufttilførsel svarende til et luftskifte på mindst 0.5 gange per time i boliger. Kravet om konstant udelufttilførsel er i ikke i overensstemmelse med det tidsvarierende behov for ventilation i boliger, der afhænger af tilstedeværelsen af personer, emissioner af forurening m.v., og resulterer i perioder med dårlig luftkvalitet og/eller unødvendigt energiforbrug. Hvis tilførslen af udeluft varieres efter behovet kan indeklimaet forbedres, og energiforbruget til ventilation kan reduceres i forhold til et system med konstant udelufttilførsel.

Et litteraturstudie af forureningskomponenter i boliger, deres kilder og deres indvirkning på mennesker dannede grundlaget for behovsspecifikationen. Emissionen af forurening i boliger består tilnærmelsesvist af konstant emitterende baggrundskilder og trinvist konstant emitterede kilder relateret til tilstedeværelsen af personer og aktiviteter. Teoretiske analyser af disse to kilder viste de luftkvalitetsmæssige konsekvenser forbundet med de tidsvarierende luftstrømme i et ventilationssystem styret efter tilstedeværelsen af personer i forhold til et system med konstant udelufttilførsel. Analyserne blev desuden brugt til at beskrive den potentielle reduktion i udskiftet luft forbundet med tilstedeværelsesbaseret behovsstyret ventilation i forhold til et system med konstant udelufttilførsel under forudsætning af, at de to systemer foranledigede den samme gennemsnitlige eksponering af forureningskomponenter. Resultaterne viste, at mængden af udskiftet luft kan reduceres med op til 26% i et tilstedeværelsesbaseret behovsstyret ventilationssystem i forhold til et system med konstant ventilation. Dette er dog på bekostning af en øget maksimal koncentration. Det behovsstyrede systems tidsvarierende luftstrømme forventes dog ikke at give anledning til akutte helbredsmæssige problemer. Design af systemer var fokuseret på simple og omkostningseffektive løsninger til et centralt balanceret behovsstyret system med varmegenvinding. Et design, der forventedes at opfylde dette krav, blev undersøgt i detaljer med hensyn til dets elforbrug ved at evaluere en reguleringsstrategi, som tilpasser setpunktet for det statiske tryk ved delbelastning. Resultaterne viste, at denne reguleringsstrategi kan reducere elforbruget med 20% til 30% i forhold til et system med konstant statisk trykregulering.

Projektet tilvejebringer fleksible metoder til design af boligventilation, der sørger for at tilstedeværelsesbaseret behovsstyret ventilation overholder bygningsreglementer og standarder, der i øjeblikket er baseret på konstant udelufttilførsel. Desuden foreslås et simpelt, omkostnings- og energieffektivt design til behovsstyret ventilation i etageboliger.

Table of Contents

Additional work (not included in the thesis)

[1] Mortensen D.K., Nielsen, T.R. & Topp, C.: 'Comparison of a constant air volume (CAV) and a demand controlled ventilation (DCV) system in a Residential building'. In proceedings: *8th Symposium on Building Physics in the Nordic Countries, 2008, Copenhagen (Denmark).*

[2] Mortensen D.K., Nielsen, T.R. & Topp, C.: 'Comparison of a constant air volume system and a demand controlled ventilation system in a residential building'. In proceedings: *11th International Conference on Indoor Air Quality and Climate, 2008, Copenhagen (Denmark).*

[3] Mortensen D.K., Nielsen, T.R. & Svendsen, S.: 'Demand controlled ventilation in a bathroom'. In proceedings: *29th AIVC Conference, 2008, Kyoto (Japan).*

[4] Mortensen D.K., Nielsen, T.R., Svendsen, S. & Christoffersen, L.D.: 'Rational use of supply air in residential buildings'. In proceedings: *9th International Conference on Healthy Buildings, 2009, Syracuse (New York, US).*

[5] Mortensen D.K., Sherman M.H. & Walker I.S.: 'IAQ based design of an efficient DCV system'. In proceedings: *10th Clima Conference, 2010, Antalya (Turkey).*

[6] Mortensen D.K., Walker, I.S. & Sherman, M.H.: 'Energy and air quality implications of passive stack ventilation in residential buildings'. In proceedings: *ACEEE, 2010, Asilomar (California, US).*

Nomenclature

A	Air change rate (or dilution rate)	h^{-1}
A	Area	m^2
d	Dose	h
C	Concentration	$-$
f	Fractional time in the period T	$-$
k	Constant depending on system design	$Pa/(m^3/s)^n$
K	Constant depending on diffusor design	$-$
LVF	Low-ventilation factor	$-$
n	Flow exponent	$-$
N	Dilution	$-$
p	Pressure	Pa
P	Power	W
q	Air flow rate	l/s
r	Fractional dilution rate in the step	$-$
S	Source strength	h^{-1}
t	Time	h
T	Cyclic period	h
v	Velocity	m/s
V	Volume	m^3
W	Weighting	$-$
x	Length	m
Z	Simplifying variable	$-$

Subscript	
B	Basic
$background$	Background sources
CAV	Constant Air Volume system
DCV	Demand Controlled Ventilation system
F	Forced
$high$	High value
L	Low
low	Low value
o	Outlet
$occupant$	Occupant related sources

max	Maximum	
min	Minimum	
r	Relative throw	
ref	Reference case	
$test$	Test case	
x	Velocity at distance x	
$*$	Reference system	
0	Intermittent system	
1	1^{st} period	
2	2^{nd} period	
Greek symbol		
ϵ	Ventilation effectiveness	$-$
η	Efficiency	$-$
ϕ	Simplifying variable	$-$
Abbreviations		
ACH	Air Changes per Hours	
AHU	Air Handling Unit	
ASHRAE	American Society of Heating, Refrigeration, and Air-Conditioning Engineers	
BRI	Building Related Illness	
CAD	Canadian Dollar	
CAV	Constant Air Volume	
DCV	Demand Controlled Ventilation	
DKK	Danish Kroner	
ER	Emission Ratio	
HAM	Heat, Air and Moisture	
IAQ	Indoor Air Quality	
IEA	International Energy Agency	
LVF	Low-ventilation factor	
MDF	Medium-density fiberboard	
occ	Occupant	
PM	Particulate Matter	
RH	Relative Humidity	
SBS	Sick Building Syndrome	
SFP	Specific Fan Power	
SPR	Static Pressure Reset	
SVOC	Semi Volatile Organic Compounds	
VAV	Variable Air Volume	
VOC	Volatile Organic Compounds	
VSD	Variable Speed Drive	
WHO	World Health Organization	

Part I

Introduction and summary

Chapter 1

Introduction

Emphasis on energy savings in buildings has to a great extent focused on improving the thermal properties of the building envelope. This has reduced the heat loss through the building envelope and the energy consumption related to operation of building services therefore accounts for a larger part of the total energy consumption of a building. This includes the ventilation system. Ventilation is strongly associated with comfort and health [1] and is a key factor in maintaining a good indoor air quality (IAQ) for the occupants. Furthermore, ventilation is also needed to prevent condensation on indoor surfaces that can cause rot and mould growth and eventually damage the building structure. Residential ventilation has traditionally been accommodated by intentional openings and infiltration through leaks in the building envelope often assisted by mechanical exhaust from bathrooms and kitchens. Air flows were partly or entirely driven by the local time-varying buoyancy and wind effects and could therefore vary significantly over time. The improved energy performance of the building envelope has made buildings more air tight and thereby eliminated several of the prevailing ventilation pathways in residences. Ventilation of tight buildings must be planned carefully and mechanical systems that control supply and exhaust air flows are often installed. Therefore, the ventilation air flows previously driven by the often uncontrollable temperature and wind effects have become more stable and controllable. The greater control of in- and outflows have prepared the ground for intelligent control of indoor air quality.

People in the developed world spend more than 90% of the time indoors [2]. Furthermore, people spend 65% of the time in their homes [3] making them a central place of exposure. The exposure depends on the indoor pollutant levels that generally are controlled by ventilation whereby polluted indoor air is exchanged with fresh outdoor air. The easiest way to express air quality requirements are through air flow rates. The Danish Building Code [4] requires a constant air flow rate equivalent to at least 0.5 air changes per hour in residential buildings. This rate has empirically proven to keep the relative humidity at a level where severe moisture related problems are avoided. However, the requirement is inconsistent with the time varying needs for ventilation of residential buildings that depend on occupancy, pollutant emission, etc., and results in periods with poor air quality and/or unnecessary energy consumption. If the ventilation rate is varied according to the need of both occupants and the building, the indoor climate can be improved and the energy consumption for conditioning and transporting the air can potentially be reduced compared to a system with constant air flow.

1.1 Aim and objective

The hypothesis of the project is that the energy consumption for ventilation in residential buildings can be reduced by demand controlled ventilation and at the same time provide the same or better indoor air quality compared to constant air flow requirements. The need for ventilation in residential buildings and the potential savings are analyzed in the project. On this basis, new energy-efficient solutions are developed based on the most important parameters identified in the demand analysis. The project focuses on solutions for multi-family dwellings. However, the results are also expected to be applicable in other types of buildings.

1.2 Thesis outline

An overview of demand controlled ventilation (DCV) with emphasis on its application in residential buildings is given in chapter 2. The overview summarizes work done in previous studies and suggestions to focus areas for future work. Furthermore, aspects on DCV system design regarding air flow requirements, energy requirements and energy-efficient air flow control strategies are given.

In chapter 3 a literature study on indoor pollutants in homes, their sources and their impact on humans are given. The chapter is completed with an overview of priority pollutants in homes based on existing literature.

Chapter 4 is initiated with a discussion of variables relevant for control of residential DCV systems. Through theoretical calculations of the behavior of these variables the air quality implications associated with DCV in comparison to constant air volume (CAV) systems were analyzed. These analyses were also used to optimize air flow rates and through that describe the potential air flow savings associated with DCV systems that provide the same average occupant exposure as a CAV system.

On this basis simple and cost-effective system designs for centrally balanced DCV systems were developed and the work is summarized in chapter 5. A design expected to meet the requirement was investigated in detail with regard to its electricity consumption by evaluation of different air flow control strategies.

Finally a conclusion is given in chapter 6 together with suggestions for future work.

Chapter 2

Demand controlled ventilation systems

Ventilation is principally used to maintain acceptable indoor air quality by controlling indoor pollutant concentrations and minimizing occupant exposures to the pollutants. Besides the atmospheric environment also the thermal and acoustic environment is affected by ventilation through draught, noise, etc. Unavoidable and non-specific pollutants in the indoor air are normally diluted with outdoor air (presumably fresh and clean) to lower concentrations whereas specific pollutants are best dealt with by direct source control measures such as local exhaust. Though ventilation is used to control pollutant concentrations, pollutants are best addressed by removing their sources from the indoor environment when possible.

The process of exchanging air in buildings can be accomplished by different types of systems. The types are characterized by the ventilation principle and the forces used to exchange the air. The three main principles in Denmark are: natural ventilation, exhaust ventilation and balanced ventilation. Ventilation driven by wind and buoyancy effects is denoted as natural ventilation and exploits these forces to exchange air often through a stack. For mechanical ventilation a fan is used to drive the process. Mechanical ventilation can work to exhaust air and thereby increase in-flows elsewhere in the building envelope or balance exhaust and supply air flows. Application of mechanical ventilation ensures a minimum exchange of air in the building.

2.1 Demand Controlled Ventilation

Demand controlled ventilation is a method to maintain a certain indoor environmental quality by adjusting the outdoor air flow according to a measured demand indicator. 'Demand' can refer to criteria related to atmospheric and thermal climate. However, in this work demand only refers to factors affecting air quality and not thermal comfort. The interest in DCV has been ongoing through the past decades. From 1987 to 1992 the International Energy Agency's (IEA) programme on Energy Conservation in Buildings and Community Systems (ECBCS) carried out a research project on demand controlled ventilation labelled Annex 18 [5–10]. The objective of the annex was to develop actions, methods and strategies for efficient DCV systems in residential, office and school buildings. Numerous projects have since expanded the knowledge of the DCV concept in both commercial and residential buildings.

The air flow rate in DCV systems is determined by sensors detecting pollutants such as CO_2, humidity, VOC etc. or can be set by simple timers, presence detection sensors etc. Systems using control variables that detect the presence of load to control the air flow (feed-forward control) do not provide information on the state of the room air quality and are therefore typically not as energy-efficient as systems using control variables detecting the magnitude of a pollutant to control the air flow (feed-back control)[11]. This relationship is illustrated in figure 2.1 where the energy performance of various DCV strategies are ranked. The figure is adopted from Wouters [12]. The assumed energy performance improvement (denoted real performance improvement) is given on the horizontal scale and shows that presence detection probably will have an improved energy performance compared to manual control.

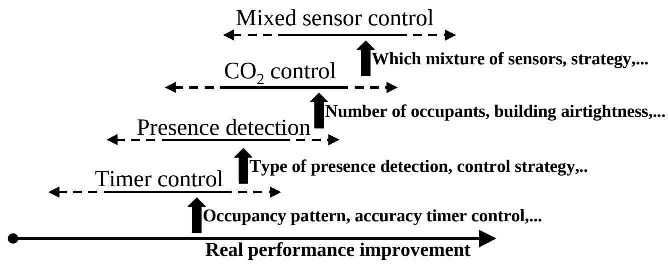

Figure 2.1: Various strategies of demand controlled ventilation have different impacts on the energy use. The figure is adopted from Wouters [12].

However, independent of the control variable the rationale in controlling the outdoor air flow according to the demand is that air is used when needed. Periods with over-ventilation and associated energy penalty or under-ventilation and poor air quality are avoided. DCV systems ideally intend to maintain a certain indoor air quality with a minimum of purchased energy.

2.1.1 DCV strategies in residential buildings

Several IAQ control variables have been used to determine the demand for ventilation. The typical variables used to control DCV systems in residential buildings are:

- Humidity
- CO_2
- Occupancy

As mentioned, the control variable can also refer to demand related to thermal comfort, typically by room temperature. Such systems are primarily known from commercial buildings where the heat load varies during the day due to changes in occupant density, equipment use, outdoor weather conditions etc. A Norwegian study on balanced mechanical ventilation in apartment buildings found that problems with overheating never had been a problem with exhaust ventilation systems, but that complaints about overheating occurred in new apartment buildings with balanced ventilation [13]. One reason was that the temperature increased in the supply duct. A detailed investigation of heat gains in duct systems have been made by Maripuu [14]. The issue of overheating in residential buildings is assumed to be of modest concern in a northern European context. Possible concerns should be dealt with in the design phase by identification of the reasons (heat gain in ducts, high internal loads) and possible ways to manage it (insulate ventilation ducts, window airing, shading) rather than controlling the ventilation air flow by room temperature. However, it is important to keep in mind that thermal comfort is ranked by building occupants to be of greater importance compared with visual and acoustic comfort and good air quality [15].

In the following paragraphs the most common control variables for DCV ventilation in residential buildings are summarized from literature and experience with their performance is given.

CO_2 is found to be a good indicator of occupancy as it appears to track human presence and activities well [16, 17]. However, CO_2 as a control variable is only appropriate if occupancy is not very low and is likely to be most effective with unpredictable variations in occupancy [17, 18]. Furthermore, CO_2 can control occupant generated pollutants effectively but may cause problems with other pollutants that are not associated with occupancy [18, 19]. A study that used CO_2 as the only control variable in a dwelling found an increased risk of high humidity and therefore recommended combining CO_2 detection with humidity detection [20].

Humidity is an attractive control variable in residential buildings because it is directly related to the parameter it is intended to manage. This includes moisture generated by activities such as showering and cooking. The part of Annex 18 dealing with residential buildings concluded that moisture usually is the dominant pollutant in these premises [10]. Relative Humidity (RH), that is the fraction of water vapor content in the air at a given temperature relative to the water vapor content in saturated air at the same temperature, is the most common humidity parameter to measure. Humidity can also be expressed solely by the content of water vapor in the air as absolute humidity. Several authors have argued and shown that relative humidity is poorly correlated with occupancy and therefore do not reflect the level of human occupancy accurately [9, 16, 19, 21]. This is because RH depends on temperature and because of absorption and desorption of moisture by building materials and furniture in the home. A DCV system based on relative humidity as the only control variable showed increased risk for poor air quality (high CO_2 concentration) when the home was occupied [20]. The correlation between absolute humidity and CO_2 concentration is stronger than the correlation between relative humidity and CO_2 concentration. However, absolute humidity still displays a lag time due to the sorption characteristic of the house [21, 22]. The effect of hygroscopic buffering in a bedroom was investigated for different air flow rates and it was found that the indoor humidity conditions fluctuated less when the hygroscopic properties of building materials were included compared to when they were not included. The building materials effect on the relative humidity was most pronounced at low air change rates (0.1 h^{-1}) compared high air change rates (1.0 h^{-1}) [23]. Other aspects of hygroscopic buffering were investigated as part of the IEA Annex 41 on Whole building Heat, Air and Moisture Response [24]. This included comparison of four Heat, Air and Moisture (HAM) simulation tools evaluating the performance of a RH based residential DCV system [25]. In addition to the internal production of moisture, the ventilation rate and hygroscopic buffering, the outdoor climate impacts the humidity level. The water content of outside air is higher during the summer season in Denmark and the indoor relative humidity therefore increases. The air flow in DCV systems with a fixed set point of a RH sensor therefore increases unnecessarily during this period. Several studies have reported this [25, 26]. The problem can be solved if the set point is changed during the year [9].

Occupancy detection as a control variable is not directly related to the quality of the air but to the presence of load and guarantee higher air flows when occupant are present. A study using occupancy as a control variable in a home found that it has a poor short term correlation with CO_2 concentration but excellent long term correlation [21]. In another study a similar finding was reported but also that occupancy detection as a control variable must be combined with a control variable sensing humidity to give a satisfactory indoor climate [20]. Generally it is stated that occupancy detection is an appropriate control variable in buildings where the number of occupants during the period of occupancy is relatively stable and may provide the most cost-effective solution for sensor based DCV [19].

Other control possibilities for DCV include time control where the ventilation system is operated by a simple clock that is adjusted to the occupant schedule or manual control where the ventilation rate is controlled by the occupant. Manual control is typically used for operation of range hoods in kitchens. The report "*The state-of-the-art in sensor technology for Demand Controlled Ventilation*" published in 2005 provides an extensive look upon available sensors, their performance and cost by review of literature on the latest technology and a market survey on cost [27]. According to the report the cost of CO_2 sensors were approximately \$500 CAD (∼2500 DKK) and the cost of RH sensors depending on technology was below this. The current cost of a CO_2 sensor is approximately 3000 DKK excluding VAT [28]. Presence detectors are sold in most builders merchants and in online stores for approximately 400 to 800 DKK [29].

2.1.2 Evaluation of DCV in residential building

The performance of a DCV system is evaluated on how well it provides an acceptable air quality and how much energy it consumes. CO_2 and RH levels are used in some studies to evaluate the air quality in homes [20, 30, 31]. However, several studies and literature reviews state that this is insufficient to determine if time-varying ventilation rates cause problems with other pollutants e.g. non-occupant related background pollutants such as formaldehyde [19, 25, 27]. The periods where the DCV system is operated at reduced air flow rates will increase the concentration of these pollutants. Reducing the ventilation rate and thereby the energy used for ventilation should be made without compromising the air quality. A DCV systems lower and upper air flow rates should therefore be carefully chosen [25].

Ventilation of buildings requires energy to transport and condition the air. Savings associated with DCV are typically reported as the reduced energy to condition the air. This saving can be reported in different forms such as reduced average air flow and reduced ventilation heat loss. The reported saving are highly dependent on factors such as DCV strategy (occupancy, RH, CO_2, etc.), set point of the control variable (e.g. 800 ppm, 1000 ppm etc.), occupant pattern and density (young, adult, singles, couples, families), outdoor climate (cold, moderate, mild etc.), system design (natural exhaust, exhaust system with fan, balanced), the model for simulation (1-zone, multi-zone, thermal model, HAM model), etc. Despite variation of these factors, savings are commonly reported compared to the performance of a CAV system. Table 2.1 summarizes residential DCV savings reported in the literature with indication of the control variable and whether the result is based on simulations (S), measurements(M) or is from a literature review (R).

Of the reported savings associated with DCV given in table 2.1 17 studies used humidity as a control variable, 7 used CO_2 and 6 used occupancy detection.

2.1.3 Focus areas for future work

During a workshop on Demand Controlled Ventilation at the Clima conference in 2010 the following future research topics on DCV were listed [17]:

Demand specifications need to consider safety from short-time hazards, the health effects of long-term hazards, the general well being including human performance. These aspects needs to be quantified in engineering terms and take a critical view of the use of energy.

Demand control variables with regard to thermal comfort are well known. Control variables with regard to air quality require greater understanding. The effects of gases and particles (from the building and activity) on health needs to be quantified and included as a control variable.

Demand variability through understanding occupancy patterns of various building types is key for a successful DCV strategy. The larger the variability, the greater the potential of a DCV system to reduce energy use.

DCV components generally need to be designed to handle large variations in flow. Key components requiring research are fans, diffusers, and sensors.

DCV system design in general is needed to continue to develop improvements. A key focus area is individual control of IAQ providing customer satisfaction. Future designs need to have greater flexibility to changes in conditions.

The work in this thesis deals with the topics of *demand specification* and *system design* for residential buildings.

Table 2.1: Publications on residential DCV

Publication	Control variable	Average air flow reduction	Reported savings (heat)	Study type[a]
Parekh et al. [16] (1991)	RH	8%		M
Boligministeriet [32] (1992)	RH	27%		M
Kesselring et al. [33](1993)	CO_2		-15% to -20% [b]	M
Mansson et al. [9](1993)	Manual, moisture		5% to 15%	R
Heinonen et al. [34](1994)	CO_2, RH	40%	40%	M
Nielsen et al. [35](1995)	RH	28%		M
Bergsøe [36] (2000)	RH	20% to 30%		S
Römer [37](2001)	Occupancy, RH		15% to 20%[c]	S
Pavlovas [20](2003)	RH		60%	S
Pavlovas [20](2003)	CO_2		>65%	S
Pavlovas [20](2003)	Occupancy		20%	S
Afshari et al. [38] (2005)	RH	31%		M
Pavlovas [39](2006)	Occupancy, RH		5%	S/M[d]
den Bossche [31] (2007)	Occupancy, RH		14% to 27%[e]	S
Bergsøe et al. [40] (2008)	RH	- 20%[f]		M
Woloszyn et al. [25] (2008)	RH	30% to 40%	12% to 17%[g]	S
Nielsen et al. [30](2009)	Abs. hum, CO_2	20%		S
Nielsen et al. [41](2010)	Abs. hum, CO_2	23%		M
Laverge et al. [42](2011)	RH		25%	S
Laverge et al. [42](2011)	Occupancy		25%	S
Laverge et al. [42](2011)	CO_2		25%	S
Laverge et al. [42](2011)	RH, occupancy, CO_2		60%	S

[a]S=simulation, M=measurements, R=literature review
[b]Saving is negative because it was compared to a system with a low air change rate of 0.1 h^{-1}
[c]Saving depended on the setting for occupancy detection
[d]Saving was calculated partly from measurement and partly from simulation result
[e]Saving depended on air tightness of the building envelope
[f]Saving is negative because it was compared to a system with a ventilation rate lower than required
[g]Saving depended on used simulation tool

2.2 Aspects on DCV system design

In the following section aspects related to demand specification and system design of DCV for residential buildings are considered.

2.2.1 Air flow requirements

Most standards and building codes specify desired levels of indoor air quality through ventilation rate requirements assuming typical emission of pollutants. This prescriptive approach is appealing because designers and engineers to a large extent avoid to deal with the unknowable exposure occupants are subjected to and because the approach provides simple and easily applicable requirements in terms of solutions [43]. However, ventilation is not an end in itself but is part of the system intended to provide a desired level of indoor air quality. Contrary to the prescriptive approach, a performance oriented approach focus on the desired level of indoor air quality. The prescriptive approach specify requirements in terms of solutions whereas a performance based approach specify requirements in terms of pollutant concentrations and/or occupant exposures [43]. Residential building codes and standards in Europe and US have included or are considering to include performance oriented approaches as an alternative to the prescriptive requirements [12, 44]. This will allow engineers to develop innovative solutions and apply new technologies to improve both air quality and energy-efficiency [44]. Solutions that combine the performance based approach with prescriptive requirements also exist. Standard EN15251 specify 'unoccupied periods' in residential buildings as periods where there is no demand and recommend a minimum ventilation rate between 0.05 to 0.1 l/s m^2 during this period if no value is given at national level [45]. This minimum ventilation rate corresponds to an air change rate of 0.07 to 0.14 h^{-1} in a home with a typical room height of 2.5 m. The Swedish building code allow a minimum ventilation rate of 0.1 l/s m^2 when occupants are absent and 0.35 l/s m^2 when occupants are present [46].

When assessing or developing innovative ventilation systems that are not covered by the prescriptive approaches in codes and standards the *principle of equivalency* can be used to show compliance. This means that one has to show that the same (or better) performance as that obtained by following codes and standards can be achieved [12]. The performance of a system designed according to prescriptive requirements sets the target for the new system. The performance target of a ventilation system can be e.g. equivalent air quality, equivalent energy-efficiency, etc. Even though one parameter sets the target for equivalent performance the others must be evaluated to reflect the performance of the system as a whole. Standard EN 15665 [47] suggests several air quality criteria to evaluate the performance of residential ventilation systems. These cover: threshold of the pollutant concentration, weighted average concentration, average concentration above a threshold with limited compensation, average concentration above a limit and dose above a given value or decay criterion. The chosen criterion depend on the key pollutant that is defined as the most important pollutant in the considered space.

The principle of equivalency assumes that current air flow requirements in standards and codes produce acceptable indoor air quality. However, there is often little knowledge about the basis for these requirements. The project HealthVent (2010-2012) supported by EU's Executive Agency for Health and Consumers (EAHC) is currently working to increase the knowledge in this area [48]. The aim of the project is to develop health-based ventilation guidelines reconciling health and energy in offices, homes and public buildings such as schools, nurseries and daycare centers. The project is carried out by a combination of experts in medicine, engineering, indoor air sciences, exposure assessment, energy evaluation and ventilation practices

The principle of equivalency has been used to assess the performance of a hybrid ventilation system at a Dutch school [12] but also to specify prescriptive requirements for the operation of intermittent ventilation systems[1] [49]. Sherman and Wilson used a procedure similar to the principle of equivalency to specify how well time-varying infiltration rates control indoor air quality [50]. They determined the constant air flow rate that yield

[1]Intermittent ventilation means that the air flow is operated on an on/off basis

the same average concentration as the actual time-varying infiltration assuming constant emission of indoor pollutants. From this they defined the (temporal) ventilation effectiveness as a measure of how well the time-varying air flow conditions control indoor air quality. The same method was used to set ventilation rate requirements for intermittent systems. The average concentration was used as the equivalency metric. The results have been included in ASHRAE Standard 62.2 by allowing intermittent ventilation provided that the ventilation rate is increased when the ventilation system is operated [51].

2.2.2 Energy requirements

The Danish Building code specifies that the total energy consumption in a dwelling must not exceed the value calculated by equation 2.1 where A is the heated floor area [4]. The total energy consumption of a dwelling is calculated as the energy used for heating, cooling, domestic hot water and ventilation where the consumption of electricity is multiplied by a site-to-source[2] factor of 2.5.

$$\text{Energy frame} = (52.5 + \frac{1650}{A})\text{kWh/m}^2 \text{ per year} \tag{2.1}$$

A dwelling can be classified as a low-energy building class 2015 if its energy consumption does not exceed the value calculated by equation 2.2.

$$\text{Low-energy class 2015} = (30 + \frac{1000}{A})\text{kWh/m}^2 \text{ per year} \tag{2.2}$$

In 2020 the total energy requirement to a low-energy class dwelling is expected to be approximately 20 kWh/m^2. These requirements must be kept in mind in the design phase of a ventilation system. In addition to the requirement to the total energy consumption of a dwelling the following paragraphs specify some general requirements to ventilation systems:

"*Ventilation systems must be carried out properly on the basis of safety, energy and indoor climate considerations*" and "*Ventilation systems must be designed, commissioned, operated and maintained so that they during the time of use at least provide the intended benefits*" [4].

The energy consumption of a ventilation system is the sum of energy needed to condition and transport the air. This consumption can account for up to half the total energy consumption in dwellings. In cold/moderate climates the energy consumption for heating the supply air accounts for the largest part. This energy consumption can be reduced by application of a heat recovery unit and/or implementation of demand controlled ventilation. If the total energy consumption of the building is required to be very low it may be necessary to include both measures. The annual energy consumption for ventilation can be estimated by a simple energy calculation based on heating degree days and system properties. Table 2.2 shows the estimated annual heating and electricity consumption related to ventilation for a CAV system without heat recovery, a CAV system with heat recovery and a DCV system with heat recovery. The estimation was made for a dwelling of 70 m^2 with an air change rate of 0.5 h^{-1} before DCV is implemented. The CAV system

[2]Site-to-source factor is also known as the primary energy conversion factor

without heat recovery has a specific fan power (SFP) value of 1000 J/m^3. The CAV system with heat recovery has a SFP value of 1200 J/m^3 and the heat recovery unit has an efficiency of 80%. The energy consumption for heating the ventilation air is calculated for a climate with 3000 heating degree days approximately equivalent to the Danish climate. Heating and electricity savings associated with DCV are assumed to be 30% of the CAV system with heat recovery.

Table 2.2: Estimated annual heating and electricity consumption of three ventilation systems: A CAV system without heat recovery, a CAV system with heat recovery and a DCV system with heat recovery. The last column show the total energy consumption per m^2 when electricity is multiplied by a site-to-source factor of 2.5.

70 m^2 dwelling	Annual heating consumption	Annual electricity consumption	Total energy consumption
CAV without heat recovery	2120 kWh	213 kWh	37.9 kWh/m^2
CAV with heat recovery	424 kWh	256 kWh	15.2 kWh/m^2
DCV with heat recovery	297 kWh	179 kWh	10.6 kWh/m^2

In case the ventilation system is combined with heat recovery, the electricity used to transport the air will take up a larger part of the total energy consumption of the system compared to a system without heat recovery. To further reduce the energy consumption of the ventilation system the target would be to reduce the fan electricity consumption by using good duct design, proper fan selection and optimizing the control strategy.

Several authors have reported that the payback time for DCV systems with heat recovery is poor [19, 42]. This was also demonstrated in Paper III where the net present value of the additional savings by implementing DCV in a ventilation system with an efficient heat exchanger was estimated to 3400 DKK per dwelling during the life time of the system. The net present value was estimated by the savings given in table 2.2. It is therefore essential that the DCV system is simple to be cost-effective when implemented in a ventilation system with heat recovery. Overall, energy-efficient residential ventilation must be "*economically viable, their installation cost must not be critical and the maintenance should be the lowest and the most simple, and the performance must not suffer from the weight of using years*" [22].

2.2.3 Energy-efficient strategy for air flow control

Ventilation systems in multi-family dwellings, which are the focus of this work, are often centralized and air flow control strategies are needed to distribute the air where needed. Because the fundamental function of a DCV system is to vary the air flow it is natural to look at operation and control of Variable Air Volume (VAV) systems. Local control components i.e. dampers and/or air terminals vary the flow by decreasing or increasing the mechanical energy losses along the flow path. To provide stability and ensure correct distribution of the air, the static pressure is usually maintained at a fixed level at a selected point in the main duct. It is recommended to locate the sensor 75% to 100% of the distance from the first to the most remote terminal [52]. The most energy-efficient way to maintain the static pressure at the fixed set point is to alter the speed of the fan by a

Variable Speed Drive (VSD) rater than using a damper (throttling) to maintain the fixed static pressure at a constant fan speed. The pathways for the two control strategies are seen in left and middle illustration in figure 2.2. However, the full energy saving potential is not obtained because pressure must be throttled off at part load conditions to maintain the static pressure at the fixed set point. The degree of throttling in a ventilation system should be as low as possible to reduce fan energy consumption and avoid problems with noise [53]. Resetting the static pressure set point at part load reduces throttling and energy can be saved, see right illustration in figure 2.2.

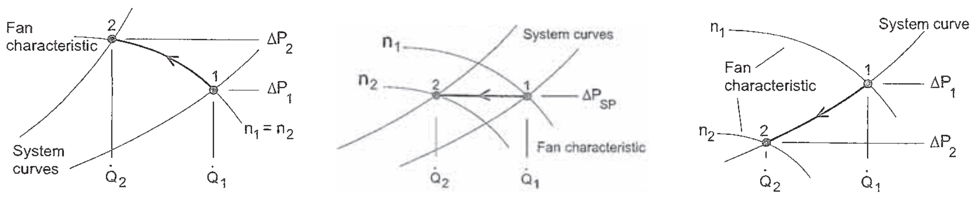

Figure 2.2: Fan diagram paths from one air flow rate to a lower air flow rate. Left figure: Damper control and constant fan speed. Middle figure: Constant static pressure and variable fan speed. Right figure: Reset of static pressure and variable fan speed. The figures are adopted from Sørensen [53]

The reset value should be high enough to avoid that zones are starved of air and low enough to keep the damper along the critical path fully open to avoid throttling. The critical path in a system with variable air flow rates continually changes as loads in a building change [52] and a static pressure reset (SPR) control method is needed to identify these changes. This control strategy is typically known from VAV systems in commercial buildings. However, it can provide energy savings in any ventilation system where the air flow varies. The strategy can be applied by different methods. Some methods use control component position or a saturation signal (derived from e.g. the air flow through or the position of the local control component) to generate a pressure request while others use predefined empirical reset schedules or a calibrated model of the ventilation system to make instantaneous calculations of the critical pressure [54]. The saturation signal method can use dampers where the actual position is unknown but only indicates if the damper is full open. Fan energy savings by resetting the static pressure range from 30% to 50% compared to fixed pressure set point control [54].

Chapter 3

Indoor pollutants

When dealing with demand specification for DCV in residential buildings one must be familiar with the substances in the indoor environment. In this chapter indoor pollutants in homes, their sources and their impact on humans are summarized from literature.

A multidisciplinary group of scientist with expertise in medicine, epidemiology, toxicology and engineering concluded, based on a review of literature, that ventilation is strongly associated with human comfort and health [1]. The literature also indicated that there is an association between ventilation rate and productivity. Air quality affects people's comfort and health. Comfort is related to people's perception of the air and is influenced by the odors and irritants in the air. The air may be perceived as stuffy, stale, smelly, damp, irritating etc. but is considered satisfactory if the great majority of people, on entry into the room, perceive the air as acceptable [55]. The effect of air quality on people's health can be short-term acute like fatigue, burning eyes or long-term chronic effects like asthma, allergy, and cancer [56]. Chronic health effects caused by poor indoor air quality in non-industrial environments are complex and difficult to identify because they can take decades to appear [56]. Sick Building Syndrome (SBS) is a term used to define acute comfort and health problems associated with the time spent in a building but where further identification of the cause cannot determined. When the cause can be identified the term Building Related Illness (BRI) is used [57].

3.1 Indoor pollutants in homes

Pollutants in the air comprise of particles and gaseous contaminants. These substances are either brought into the indoor space from outside or they are generated indoors. Pollutants generated internally in the home originate from occupants, building materials, furnishing and the activities carried out inside the home. Outdoor sources include organic matter, pollutants from combustion processes and vehicles, bioaerosols, pollutants in the ground, etc. Moisture and bioeffluents represent two types of pollutant always present in residential buildings and are therefore summarized in separate sections.

Moisture itself has little impact on people and humans usually cannot distinguish relative humidity levels between 15% to 75% [9]. However, the effects of high or low relative humidity levels can affect peoples' comfort and health indirectly e.g.

through propagating mould and dust mite growth. Moisture in a home originates from the outdoor air, people, pets, plants and activities such as cooking, showering, cleaning, clothes drying, etc. It can also be caused by penetration of rain or snow through the building envelope or leaky plumbing. Several studies have investigated the production of moisture from people and activities in a home and a commonly used moisture production rate per person is 2.5 kg/day. The emission rate from an adult is approximately 50 g/h [56]. Figure 3.1 show moisture production rates for a family of four summarized from literature [24, 56, 58–62].

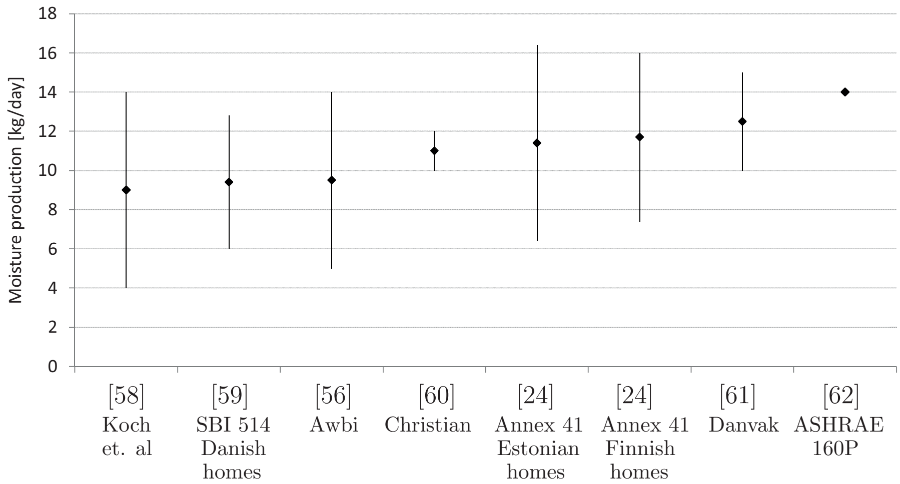

Figure 3.1: Reported moisture production rates in a home with two adults and two children. The diamonds mark the average moisture production and the bars indicate the range.

Figure 3.1 shows that literature state different moisture production rates but more importantly most literature report that the production of moisture can vary considerably from home to home. This means that the ventilation rate required to remove moisture varies from home to home. Indicators of high humidity levels are visible mould, damp stains and condensation on window panes. Relative humidity levels above 70% provide conditions for mould growth in buildings [63]. High humidity levels stimulate the growth of house dust mites and may in the long term damage the building structure [55]. Several studies have investigated associations between human health, the occurrence of mould and house dust mites and dampness. Based on an interdisciplinary literature review on the association between exposure to dampness in buildings and health effects, it was concluded that the evidence for an association between dampness and health effects is strong [64]. The review group found that dampness appears to increase the risk for a number of health effects mainly respiratory symptoms, but also unspecific symptoms like tiredness and headaches. However, it is not known which agents in the indoor air (except for mite-exposure) cause the health effects. House dust mites contain allergens that can cause allergy in humans. To keep the number of house dust mites from increasing greatly during humid parts of the year it is recommend to keep the relative humidity below 45% at 20°C to 22°C for 1 or 2 months during winter [65].

Gaseous pollutants include gases, vapours of liquids, VOCs and inorganic air pollutants. In homes these types of pollutants originate from building materials in the form of formaldehyde (CH_2O), from detergents in the form of limonene, from flooring materials and toys in the form of phthalates, from outdoor air and human respiration in the form of CO_2 (outdoor CO_2 level is around 360ppm), from the ground in the form of radon, etc. Gaseous pollutants in the indoor air are usually diluted to lower concentration by ventilation because of their non-specific distribution. Many chemicals found indoors today are different from those found decades ago and the health risks therefore also differs [66]. The occurrence and effects of VOCs and SVOCs (Semi Volatile Organic Compounds) in the indoor environment receive increased attention today due to health concerns. They have been associated with allergic symptoms in children, retardant male reproductive development, endocrine disruption etc. [67]. VOCs are organic carbon based compounds that evaporate into the air under normal atmospheric conditions and their emission rates decrease sharply during the first weeks or months of a products life. SVOCs, such as flame retardants used in beds or PCB (Polychlorinated biphenyl) previously used in building sealants [68], differ from VOCs as their emission rates tend to depend on the external environment and sorption onto indoor surfaces or airborne particles. Therefore their emission is more uniform with time and they can persist in the indoor environment for years after they have been introduced [67]. When the SVOC is airborne, either in the gas phase or absorbed to airborne particles, it can be diluted by ventilation. Formaldehyde has been a VOC of concern in homes in recent years. It is used in the adhesives typically used in plywood and carpets. There is sufficient evidence that formaldehyde is carcinogenic to humans [69]. Other symptoms in humans reported by WHO (World Health Organization) are sensory irritation to the eyes and upper airways, lung effects (asthma and allergy), eczema and besides this its odor may cause discomfort [70, 71]. The WHO recommend keeping the concentration of formaldehyde below 0.1 mg/m^3 as a 30-minute average to prevent significant sensory irritation in the general population. Recent measurements of formaldehyde concentrations in 20 newly built Danish homes (built 2001 to 2007) showed a tendency of higher formaldehyde concentrations in the newest and biggest homes [72]. The median indoor formaldehyde concentration in the homes was 0.04 mg/m^3 (range of 0.018 to 0.110 mg/m^3). Recent measurements of formaldehyde in 105 new homes (built after 2002) in California showed a median value of 0.036 mg/m^3 (range of 0.0048 to 0.136 mg/m^3) [73]. As a follow-up on the Danish measurements of formaldehyde in newly built homes the emission of formaldehyde from 22 buildings materials (wood boards, insulation materials, carpets, textiles, paints and detergents) was analyzed in a laboratory [74]. Medium-density fiberboard (MDF) was identified as the largest individual source of formaldehyde, however, emission rates did not exceed the limits set in Denmark and the EU for formaldehyde emis-

sion from wood boards.
Quality control and labeling schemes have enlightened the area related to emission of pollutants from building materials [75]. However, the labeling systems are diverse and may lead to labels having reduced efficacy. In a report by Willem and Singer that provide an overview of the current state of information available on chemical emissions of residential materials and products it is stated that there exist more than 10 labeling systems in Europe [76].

Particles are solid or liquid matter in the air [52]. Dust, exhaust from vehicles, bioaerosols (such as viruses, pollen, bacteria and fungi) can be brought into the home by the supply air and through openings in the building envelope. Particles inside the home originate from human and pet dander, fibrous material from textiles, cooking/frying, candles, smoking, wood stoves, house dust mites, etc. The reaction between some chemicals and ozone can also produce particles [77]. The concentration of airborne particles is influenced by activities and air flows in a room [75]. Resuspension[1] of PM_{10} particles (particulate matter with a aerodynamic diameter less than 10 μm) was found to be significant during vacuuming, while vacuuming had an insignificant impact on the resuspension of $PM_{2.5}$ particles [78]. Small particulate matter ($<PM_{2.5}$) are thought to constitute the greatest long-term health risk because they can accumulate and penetrate deep into the lungs [56, 75]. The European project EUROPART consisting of an interdisciplinary group of researchers reviewed literature on health effects of particles and aimed at reaching a scientific consensus about risk indicators (such as particle mass, surface area or number concentration) for health effects in non-industrial buildings [79]. The reviewed literature did not include dermal effects or cancer risks, nor studies addressing health effects and any of the following factors: tobacco smoke, house dust mites, cockroach and animal allergens, particles of microbial origin, pesticides, particles from indoor burning of biofuels. The group found that there was a fundamental lack of conclusive research in this area and therefore concluded that the scientific evidence is inadequate to establish risk indicators of health effects due to particles in non-industrial indoor environments.

Bioeffluents are the waste products of metabolism and consist of moisture, particles and gaseous contaminants. The moist sources of bioeffluents include human respiration, sweat and urine. Particles include skin fragments, and gaseous contaminants include human respiration in the form of CO_2, foul breath and gases from the digestive tract [80]. The CO_2 production by human respiration depends on the metabolic rate - an average sedentary adult produce 18 l CO_2 per hour. Bioeffluents do not result in severe health problems but CO_2 concentrations above 1000 ppm can cause discomfort and headaches [75]. Bioeffluents affect people's comfort due to release of odors from the body that depends on human's diet, activity level and personal hygiene. CO_2 cannot be filtered or absorbed and is therefore a good surrogate for human presence. Experimental studies investigating the ventilation rate required to obtain an acceptable perceived air quality have shown that a ventilation rate of

[1] Resuspension means that particles are remixed into the air

7-8 l/s per person is needed [80, 81].

3.2 Priority pollutants in indoor environments

In literature, lists have been established for priority pollutants in indoor environments. Table 3.1 is generated from three publications and show indoor pollutants identified as hazardous to humans. The European project INDEX (2002-2004) ranked indoor chemical pollutants based on their health risk and gave recommendations on how to manage those risks [82]. The highest priority was given to the following 5 pollutants because they were considered bearing to be high health risks in the European population: formaldehyde, nitrogen dioxide, carbon monoxide, benzene and naphthalene. Management options included restricting, banning and discouraging the use of the pollutants together with inspection strategies, alarm systems and changes to building codes, and ventilation and equipment/appliance standards.

A literature review by Logue of studies with reported measurements of chemical pollutants in residences was used to compile representative mid-range and upper-bound concentrations [83]. The results were compared to available health standards and 23 pollutants were identified as chronic hazards in most or some homes. Of these the following 9 pollutants were identified as priority pollutants: acetaldehyde; acrolein; benzene; 1,3-butadiene; 1,4-dichlorobenzene; formaldehyde; naphthalene; nitrogen dioxide; and $PM_{2.5}$.

In 2010 WHO published a book on guidelines for the protection of public health from 9 chemicals commonly present in indoor air and often in concentrations of concern to health [71]. The 9 substances are: benzene; carbon monoxide; formaldehyde; naphthalene; nitrogen dioxide; polycyclic aromatic hydrocarbons (especially benzo[*a*]pyrene); radon; trichloroethylene and tetrachloroethylene [71].

Table 3.1: Priority pollutants in indoor environments

Substance	INDEX [82]	Logue et al. [83]	WHO et al. [71]
acetaldehyde		x	
acrolein		x	
benzene	x	x	x
carbon monoxide	x		x
Formaldehyde	x	x	x
naphthalene	x	x	x
nitrogen dioxide	x	x	x
polycyclic aromatic hydrocarbons			x
radon			x
trichloroethylene			x
tetrachloroethylene			x
1,3-butadiene		x	
1,4-dichlorobenzene		x	
$PM_{2.5}$		x	

Table 3.1 shows that there is agreement between the three publications for some pol-

lutants. However, disagreements does not mean that the pollutants are of less concern, merely that the pollutants were identified on different grounds.

Chapter 4

Performance based ventilation rates

The time-varying air flow rates of a DCV system impact the air quality and the energy-efficiency of the system. The objective of this chapter is to determine optimal performance based air flow rates that ensure a desired level of air quality while minimizing energy use. The desired air quality level was determined by the principle of equivalency where one has to show that the same (or better) performance as that obtained by following codes and standards can be achieved. Results are based on theoretical analyses of the continuity equation using simplified emission profiles for pollutants in homes.

Emission of pollutants in residential buildings roughly fall into sources originating from building materials, furniture etc. with a more or less constant emission rate and sources associated with occupants and activities with a more or less step-wise constant emission profile, see figure 4.1.

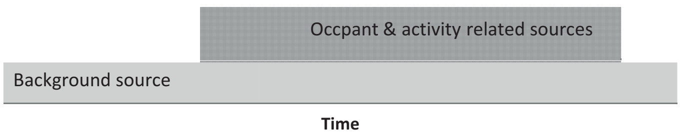

Figure 4.1: Emission profile of background and occupant & activity related sources

Several studies have demonstrated that humidity is a key pollutant in residential buildings. However, as stated by several authors, humidity is not a good indicator for occupancy. Nor does it correlate with constantly emitted background pollutants. These sources are usually dealt with by a constant ventilation rate that dilutes pollutant concentrations to lower levels. The constant rate is ideal for management of background sources, but not for the time-varying occupancy related sources. To manage the step-wise constantly emitted sources related to occupancy, a control variable that is related to occupancy is needed. CO_2 and presence sensors are typically used to detect occupancy. CO_2 as a control variable is appropriate if the occupant density is not very low and is likely to be most effective with unpredictable variations in occupancy. The occupant density in homes is lower than in offices and schools where CO_2 sensors typically are used. Moreover, the number of people living in a home is roughly constant (a 70 m^2 dwelling is typically inhabited by 1-2 persons) and variations in the presence of occupants during the day are essentially predictable - either people are at home or not. Furthermore, CO_2 sensors are expensive compared to presence sensors and will therefore have a negative effect on the cost-effectiveness of the system. Even though presence sensors rely on feed-forward control of the air flow and are not linked to the actual state of the air they are expected to be an appropriate solution with regard to air quality and cost-effectiveness in dwellings. Air quality should be evaluated by more than humidity and CO_2 levels, because these pollutants are inadequate to determine if a ventilation system causes problems with other pollutants, e.g., background pollutants with constant emission rates. Because the energy advantage of DCV systems is to reduce the ventilation rates during periods of low demand and especially when occupants are absent, the systems cause an increase in the magnitude of fluctuations in background pollutant concentrations.

The air quality implications associated with occupancy based DCV systems compared to a CAV system were investigated. The same analyses were used to optimize air flow rates and through that describe the potential air flow savings associated with occupancy based DCV systems. The analyses were based on the principle of equivalency where the DCV system provided the same air quality as a CAV system. The intent was to provide results that provide more flexible approaches to ventilation design for residences that allow occupancy based DCV approaches to comply with codes and standards that are currently based on continuous ventilation rates.

4.1 Exploring the principle of equivalency

The basis for the analyses in this chapter is the principle of equivalency. The following two aspects of the principle of equivalency were analyzed for an occupancy based DCV system using a CAV system as a reference:

- Equivalent volume of air exchanged during a day
- Equivalent dose during occupied hours

The first analysis used the volume of air exchanged during a day as the criterion for equivalent performance. The volume of exchanged air is related to the energy performance of a system and it was assumed that by exchanging the same amount of air the energy performance would remain unchanged. The air quality was therefore the parameter that could be optimized. The calculations in section 4.3 form the basis for the analysis. The peak and average concentrations were used to evaluate how the DCV system with time-varying air flow rates impact the air quality compared to a system with constant air flow rate.

The second analysis used air quality as the criterion for equivalent performance. The volume of exchanged air was therefore the parameter that could be optimized. This part is based on Paper I '*Derivation of equivalent continuous dilution for cyclic, unsteady driving forces*' and Paper II '*Optimization of occupancy based DCV systems in residences*'. The concept of dose, which is the integrated exposure to a pollutant over time, was used as the metric for equivalent air quality. It was assumed that occupant exposure and thus dose is linearly proportional to the pollutant concentration. The volume of air one would need in the system with constant air flow rate relative to that needed in the system with time-varying air flow rates was used as a measure of the systems ventilation effectiveness. Because energy consumption is related to the amount of air exchanged by a ventilation system, the ventilation effectiveness can be used as a first step to evaluate the effect of different ventilation patterns on the energy consumption.

4.2 Method

The theoretical analyses are based on the assumption that the emission of pollutants comprised of a step-wise constant part associated with occupants and a constant part emitted from building materials, furniture, etc. Short-term high polluting localized events such as cooking and showering were not specifically addressed because they often are treated separately by exhausting air from kitchens and bathrooms when these rooms are in use, although they may be considered to be part of the background and occupancy related emission over the long term.

In all analyzed cases the time at which the DCV system changed air flow rate corresponded to changes in occupancy. The occupancy based DCV system was operated at a high ventilation rate during occupied hours and at a low ventilation rate during unoccupied hours.

The indoor air quality was evaluated by the peak and average concentration of the two pollutant sources during occupied hours only. The values are intended to represent acute and chronic occupant exposures, respectively. Occupants are not exposed to pollutants when they are absent and evaluation of the peak and average concentrations were therefore limited to times when occupants were present.

One aim of the analyses was to optimize ventilation effectiveness, that is the volume of air needed in one system to that needed in another given the systems provide the same average concentration. If ventilation rate and pollutant concentration were linearly related, the average concentration would be proportional to the average ventilation rate and straightforward methods could be used to determine the effectiveness of a ventilation system with time-varying air flow rates. E.g. if the system was operated 8 hours per day, the air flow rate would need to be tripled during those 8 hours compared to the air flow in CAV system. Ventilation and concentration are non-linearly related through the continuity equation that therefore needs to be solved to determine the ventilation effectiveness.

All results are the outcome of time-varying air flow rates and not local inefficiencies associated with imperfect mixing within and between zones or the spatial distribution of pollutants in the home. Pollutants were assumed to be removed by ventilation and not by other mechanisms such as infiltration, air filtration or sorption on surfaces.

4.3 Equivalent volume of exchanged air

The indoor air quality was evaluated in a space ventilated by an occupancy based DCV system that exchanged the same total volume of air on a daily basis as a CAV system. The space was occupied 16 hours a day corresponding to the time people in general spend in their home [2, 3]. The CAV system was operated at a constant air change rate of 0.5 h^{-1} during all hours corresponding to the minimum requirement in the Danish building regulation [4]. The high and low air change rates in the occupancy based DCV system were varied, but in all cases the same total volume of air was exchanged during one day. This volume corresponded to that supplied by the CAV system. The range of possible DCV rates that supplies the same volume of air on a daily basis is limited by the low air change rate ($A_{DCV,low}$) that never can be below zero and never higher than the air change rate of the CAV system, see figure 4.2. The low air change rate was therefore used to categorize the range of DCV systems by introducing the Low-Ventilation Factor (LVF) that expressed the low air change rate as a percentage of the CAV rate, A_{CAV}, see equation 4.1. At a low-ventilation factor of 1 the low and high air change rates are identical and thereby equal to the CAV rate.

$$LVF = \frac{A_{DCV,low}}{A_{CAV}} \tag{4.1}$$

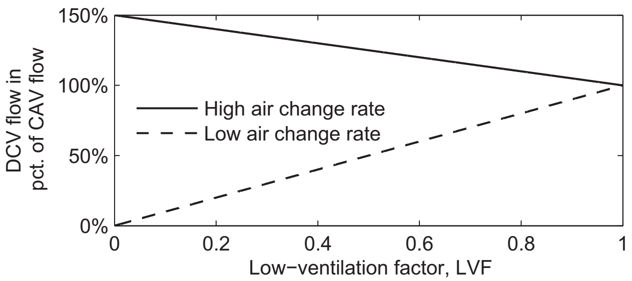

Figure 4.2: Each low-ventilation factor represents a unique DCV system with a specific high and low air change rate. All systems exchange the same total volume of air on a daily basis.

The peak and average concentrations of the two pollutant sources were evaluated separately during occupied hours. The results were normalized to the result of the CAV system. Therefore, the normalized peak and average concentrations in the CAV system (LVF=1) are unity by definition.

The daily cyclic concentration profiles of the background and occupancy related pollutants are seen in figure 4.3 for DCV systems with LVF of 0, 0.25, 0.50, 0.75 and 1. The LVF of 1 is equivalent to a CAV system. The time-varying concentration profiles were normalized to the peak concentration in the CAV system. The time of emission of the two sources is indicated by their emission profile. The first 16 hours represent the occupied period. For both pollutants it is seen that decreasing LVF results in increased concentration on entry to the enclosure at hour 0, but also a long period of time during occupancy with lower concentration. Less exchange of air during unoccupied hours reduces the dilution of pollutants during this time. Because the background source emits pollution at all times it will accumulate during the unoccupied period. Although the occupant related source has no emission at unoccupied times, a reduction of the low air change rate results in slower dilution.

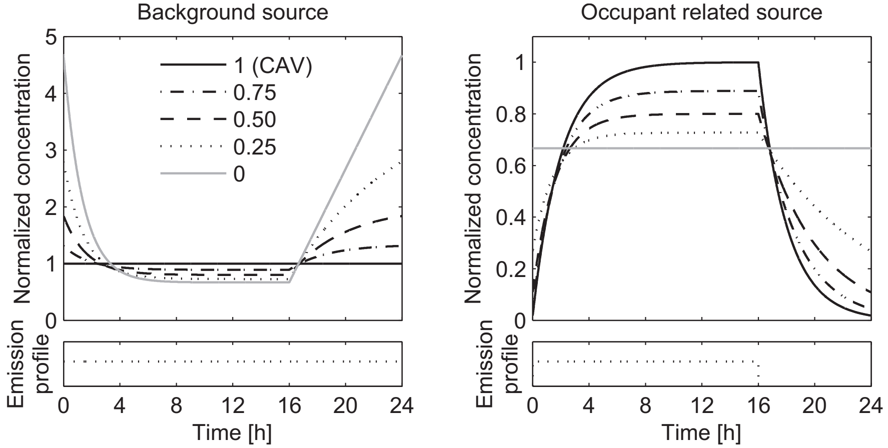

Figure 4.3: Daily cyclic concentration profiles of the background and occupancy related source for LVF of 0, 0.25, 0.50, 0.75 and 1. The time of emission for the two sources is illustrated by their emission profile. The first 16 hours represents the occupied period.

The graphs in figure 4.4 show the peak and average concentrations of the two sources for each unique DCV system represented by its LVF. The peak concentration of the occupant related source decreases as the low-ventilation factor decreases whereas the opposite applies for the background source. The highest peak concentration of the background source occurs in a system with no ventilation at unoccupied times (LVF = 0) and is approximately 4.6 times higher than that in the CAV system. At the same condition the occupancy related source has a peak concentration 0.65 times that in the CAV system.

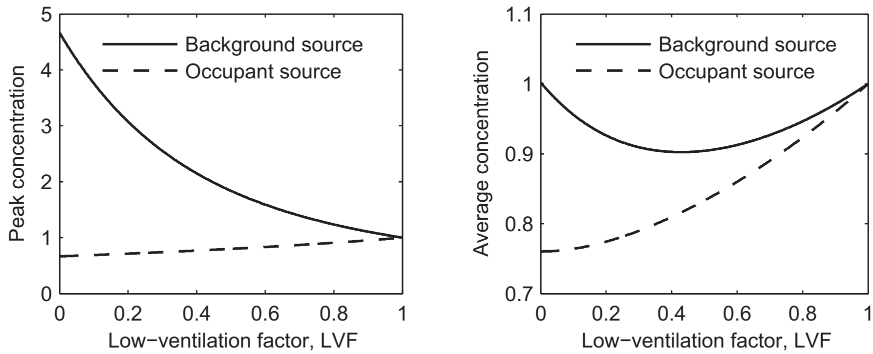

Figure 4.4: Left graph: Normalized peak concentration of the background and occupancy related sources during occupancy. Right graph: Normalized average concentrations of the background and occupancy related sources during occupancy

An increase in air change rate during times of occupancy results in a lower average concentration of the occupancy related source compared to CAV operation. In the DCV system with no ventilation during unoccupied hours (LVF = 0) the average concentration of the occupancy related source is 22% lower than that of the CAV system. The average concentration of the background source decreases with decreasing LVF until a minimum value 10% lower than in the CAV system (LVF=0.43). When LVF=0.43 the peak and average concentration of the occupancy related source is approximately 0.74 and 0.79 times that in CAV system respectively. The peak concentration of the background source is 2 times that in the CAV system. This means that the average concentration of both sources and the peak concentration of the occupancy related source can be reduced simultaneously in the occupancy controlled DCV system compared to a CAV system without exchanging more air. However, this is at the expense of increasing the peak concentration of the constantly emitted background source. The constant background source is the limiting factor for overall improvement of the air quality in the considered occupancy controlled DCV system. By careful selection of air change rates the average concentration of the background source can be reduced by 10% at the expense of doubling its peak concentration (LVF=0.43). As an example: a relatively constantly emitted pollutant source like formaldehyde with an average concentration of 0.04 mg/m^3 can in a CAV system be reduced to an average concentration of 0.036 mg/m^3 in a DCV system, but the peak concentration will increase to 0.08 mg/m^3.

Even though a normalized concentration below 1 always can be interpreted as an improvement of the air quality compared to the CAV system, values above 1 do not imply that the indoor air quality is unacceptable but only that the concentration in the DCV system exceeds that of the CAV system. Moreover, the relative importance of the background and occupancy related sources should be considered. This aspect is examined in the next section.

4.4 Equivalent dose during occupied hours

An air quality performance oriented approach was combined with the prescriptive requirements to ventilation commonly stated in building codes and standards. The air quality level obtained when meeting the constant ventilation requirements in standards and codes was used to set the target for the occupancy based DCV system such that the system would show the potential to comply with current standards and codes. The concept of dose, which is the integrated exposure to a pollutant over time, was used as the metric for equivalent air quality. It was assumed that occupant exposure and thus dose is linearly proportional to the pollutant concentration. Dose is used because the vast majority of indoor air quality issues examined for ventilation standards are limited to chronic, long-term exposure and do not address short-term acute exposures or highly toxic substances with non-linear dose response for human health.

The effectiveness of the performance oriented approach (denoted as test system) was compared to the prescriptive approach (denoted as reference system) by calculation of the ventilation effectiveness. The effectiveness is a measure of how good the test system is at providing an air quality relative to the reference system. The effectiveness is defined by the volume of air one would need in the reference system, V_{ref}, to that needed in the test system, V_{test}, over a certain period, see equation 4.2.

$$\varepsilon = \frac{\sum V_{ref}}{\sum V_{test}} \tag{4.2}$$

An effectiveness above unity means that the test system can perform better (i.e. use less air) than the reference system. The effectiveness can be used as a target or optimization parameter in the design process. The test system can be designed to match some target effectiveness that often will be unity, as the test system then exchange the same amount of air as the reference system. This situation was investigated in section 4.3. The effectiveness can also be used as an optimization parameter in designing systems that provide same dose but use less air. This was investigated in Paper II.

4.4.1 Derivation of equation for equivalent dose

In Paper I an analytical approach was used to determine the dilution of an unsteadily-generated solute in an unsteady solvent stream, under cyclic temporal boundary conditions. The goal was to find a simplified way of showing equivalence of such an unsteady test system to a reference system where equivalent dilution was defined as a weighted average concentration i.e. dose. This derivation has direct applications to the ventilation of indoor spaces where indoor air quality and energy consumption cannot in general be simultaneously optimized. By solving the equation it is possible to specify how much air is used in one ventilation pattern compared to another (i.e. the ventilation effectiveness) to provide the same indoor air quality.

The reference system was selected as the system conventionally called *perfect dilution*, which is defined as that time varying reference dilution rate, $A_*(t)$, that holds the concentration constant at some steady state value, C_*, when the time-varying source strength is $S(t)$:

$$A_*(t) = \frac{S(t)}{C_*} \tag{4.3}$$

Densities are assumed constant and equation 4.3 is normalizes by the volume of the ventilated space such that the equation is volume independent. The dose in the reference system, d_*, is given by equation 4.4 where W is the weighting function: $\oint W(t)dt = T$. The weighting function allows to emphasize parts of the cyclic period heavier than others or to omit parts of the cyclic period by using a zero value weighting factor when solving for equivalent dose e.g. during periods when occupants are not present.

$$d_* = \oint C_* W(t)dt = C_* T \tag{4.4}$$

The amount of air exchanged by the reference system, N_*, is the integrated reference dilution rate over the cyclic period:

$$N_* = \oint A_*(t)dt \tag{4.5}$$

The considered test system with the time-varying dilution rate, A, is cyclic over the period T and therefore takes on the same value when the time changes one period. The general equation for the time-varying concentration, *C(t)*, under cyclic boundary conditions is given by:

$$C(t) = \frac{\int_{t-T}^{t} S(t')\xi(t,t')dt'}{1 - \xi(T,0)} \tag{4.6}$$

Where:

$$\xi(t,t') = e^{-\int_{t'}^{t} A(u)du} \tag{4.7}$$

The derivation of the time-varying concentration is found in Paper I. The time-varying concentration was integrated over the cyclic period, T, to calculate the dose, d:

$$d = \frac{\oint W(t) \int_{t-T}^{t} S(t')\xi(t,t')dt'dt}{(1 - \xi(T,0))} \tag{4.8}$$

The amount of air exchanged by the test system, N, is:

$$N = \oint A(t)dt \tag{4.9}$$

The ventilation effectiveness is calculated by equation 4.5 and 4.9:

$$\varepsilon = \frac{\oint A_*(t)dt}{\oint A(t)dt} = \frac{N_*}{N} \tag{4.10}$$

The derived expression of the dose in the test system (eq. 4.8) can be compared to the dose in the reference system of perfect dilution (eq. 4.4). If the source strength, $S(t)$, and the time-varying dilution rate in the test system, $A(t)$, is completely specified one can determine the steady-state concentration, C_*, in the reference system of perfect dilution by first solving equation 4.8 and then 4.4. When trying to design a system that produces a dose equal to that in the reference system one can use equation 4.8 as the constraint on the test system that makes that true. The problem then reduces to finding the test dilution pattern, $A(t)$, that gives the target dose. Because the time-varying reference dilution rate, $A_*(t)$, was defined as that which holds the concentration constant at some steady state value, C_*, it is not necessary to individually know the source strength, $S(t)$, and the steady state concentration, C_*, but only the presumed dilution for perfect dilution, $A_*(t)$.

Step function

Roughly the same things occur in a home on a daily basis and it can be assumed that there is a step change in pollutant emission rates from high to low corresponding to a change from the residence being occupied to unoccupied. There is a corresponding step up at the end of the unoccupied period. The applications considered further on only involve situations in which the weighting, source strength and dilution rates are all stepwise constant with one step at time t_1, see figure 4.5. Any of the three parameters can change at the step t_1 or they must remain the same through the cyclic period, T. In other words, they have to change at the same time or not change at all.

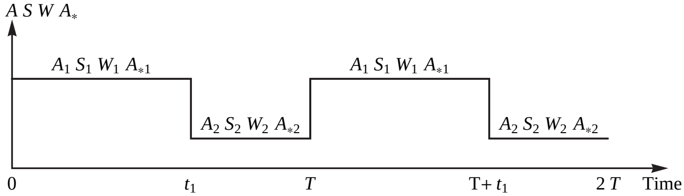

Figure 4.5: Step-wise constant weighting (W), source strength (S) and dilution rates (A, A_*) in the cyclic period T

Due to the step-wise constant profiles, the dose equation of the test system (eq. 4.8) can be expanded into a sum of integrals where the parameters are constant and an analytical expression for equivalent dose in the reference and test system can be set up (eq. 4.4 equals eq. 4.8). This analytical expression was simplified using the following definitions: weighting of the two periods: $W_1t_1 + W_2(T - t_1) = T$, reference dilution: $N_* = A_{*1}t_1 + A_{*2}(T - t_1)$, test dilution: $N = A_1t_1 + A_2(T - t_1)$, fraction of time in the step: $f = \frac{t_1}{T}$, non-dimensional test dilution rate in the step: $r = \frac{A_1T}{N}$, and non-dimensional reference dilution rate in the step: $r_* = \frac{A_{*1}T}{N_*}$. The non-dimensionalized dilution rates, r and r_*, will be 1 when the dilution rate does not change through the cyclic period and they will be 0 or T/t_1 when the dilution rate during one of the two periods is zero. The parameters r, r_* and W_1 can take on values in the interval [0, T/t_1]. By introducing the variables: $Z = fr$ and $\phi = f^2[(r - r_*)(r - W_1)]$ the equation for equivalent dilution can be reduced to:

$$N_* = \frac{N}{1 + \frac{\phi}{Z(1-Z)} - \frac{2}{N}\frac{\phi}{(Z(1-Z))^2} / \left(Coth\left[\frac{NZ}{2}\right] + Coth\left[\frac{N(1-Z)}{2}\right]\right)} \tag{4.11}$$

For detailed derivation see Paper I. ϕ and Z are only introduced to simplify the equation and allow to more easily investigate the behavior of the step-wise constant problem in its space of solutions. In equation 4.11 it is worth noting that Z is symmetrical around $\frac{1}{2}$ as replacing Z by $(1 - Z)$ yields the same result. A recursive expression for the effectiveness can be derived, but at the expense of breaking the symmetries of Z.

$$\varepsilon = \frac{1 - (\frac{\phi}{(1-Z)} - Z(1/\varepsilon - 1))(\frac{N_*}{2})(Coth\left[\frac{N_*Z}{2\varepsilon}\right] + Coth\left[\frac{N_*(1-Z)}{2\varepsilon}\right] - \frac{2\varepsilon}{N_*Z})}{1 - \frac{\phi}{(1-Z)^2}} \tag{4.12}$$

Because the considered system is step-wise constant it is only necessary to know the ratio of the reference dilution rate (A_{*1}, A_{*2}) (or source strength: S_1, S_2) in the two periods to estimate the effectiveness of the system.

Phase space of ϕ and Z

The effectiveness is a measure of how good the test system is at providing an air quality relative to the reference system. Before discussing the actual phase space of the effectiveness it is important to realize that the allowable phase space of the parameters ϕ and

Z is limited. The maximum value of ϕ occurs when the product: $(r - r_*)(r - W_1)$ is as large as possible. This occurs at two points; the first is when r_* and W_1 equals 0, hence $\phi_{max} = Z^2$. Because Z is symmetrical around $\frac{1}{2}$, $\phi_{max} = (1-Z)^2$ for Z>$\frac{1}{2}$. The minimum value of ϕ occurs when one of the differences: $(r - r_*)$ or $(r - W_1)$ is positive and the other is negative. Because r_* and W_1 take on values in the interval $[0; T/t_1]$, $\phi_{min} = Z(1-Z)$. The most negative ϕ occurs when r takes on a value exactly between r_* and W_1, hence: $r = (r_* + W_1)/2$ which means that: $\phi_{low} = -(f^2/4)(r_* - W_1)^2$. The limits of ϕ are thereby given by: $-Z(1-Z) \leq \phi \leq Maximum(Z^2, (1-Z)^2)$. The allowable phase space of ϕ and Z is given in figure 4.6.

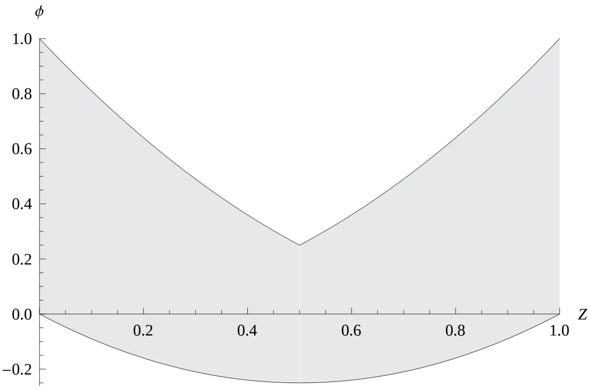

Figure 4.6: Allowable phase space of ϕ vs. Z

Intermittent dilution

In the limiting situation where there is no dilution during one of the two steps Z equals either 0 or 1. This limit is called Intermittent Dilution. The limit of equation 4.12 when Z approaches zero (or unity) results in the following expression for the effectiveness:

$$\varepsilon_0 = \frac{1}{1 - \phi + \phi(N/2)Coth\left[\frac{N}{2}\right]} \tag{4.13}$$

The subscript on the effectiveness is used to show that it is for a solution where one of the two steps has no dilution.

Phase space of effectiveness

The phase space of the effectiveness is examined by equation 4.11 and shows how good the test system is at providing an air quality relative to the reference system. An effectiveness above unity means that the test system can perform better (i.e. use less air) than the reference system. Figure 4.7 shows the dependence on ϕ for three test dilutions ($N = 2$, 5 and 10) spanning the full range of Z. For low values of test dilution, N, there is not much dependence on ϕ for the effectiveness but at higher values there is (as is there on Z). The results show that for a test dilution of two or below the effectiveness is independent of Z

and the equation for intermittent ventilation (eq. 4.13) will provide sufficiently accurate results. To design systems with an effectiveness above unity the strategy would be to minimize ϕ. If more degrees of freedom are available, Z can be optimized after that.

A reference system with constant air change rate i.e. a CAV system maintains the concentration at a steady state value, C_*, when the source is emitted constantly. To design a test system that use less air during the period T (hence, ε is above unity) the time-varying air change rates of the test system, A (represented by $r = \frac{A_1 T}{N}$) must fulfill the requirement: $r_* < r < W_1$. That way ϕ is negative. When a CAV system is the reference case, r_* is 1. In the situation where equivalent dose only is required during occupied hours, W_1 equals T/t_1. Hence, the time-varying air change rates of the test system, A, must fulfill the requirement: $1 < r < \frac{T}{t_1}$. Often it is desirable to compare a test system to a reference system that is not that of perfect dilution which holds the concentration constant at some steady state value. In such cases equation 4.8 is calculated for both systems.

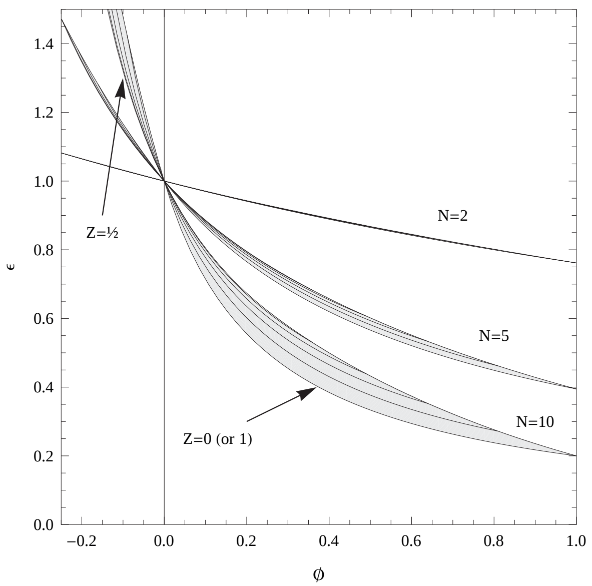

Figure 4.7: Effectiveness vs. ϕ for three different test dilutions ($N = 2, 5, 10$ from flattest to steepest). Mesh is for full range of allowable values of Z

4.4.2 Optimization of occupancy based DCV in residences

The objective of the following section is to determine the high and low air change rates of the DCV system that provide optimum ventilation effectiveness in homes. The dose equation for the test system (eq. 4.8) was used in Paper II to determine the ventilation effectiveness of DCV systems using a CAV system as the reference. The reference system is no longer the case of perfect dilution that holds the concentration constant at some steady state value as defined in section 4.4.1. The intent of the study was to provide results that can provide more flexible approaches to ventilation design that allow demand controlled ventilation approaches to comply with codes and standards that are currently based on continuous ventilation rates by showing equivalency in terms of dose. The DCV approach used occupancy as an indicator for increased demand and equivalent dose was only required during occupied hours. The occupancy based DCV system had a high air change rate during occupied hours and a low air change rate during unoccupied hours. The low-ventilation factor, LVF, given by equation 4.1 was used to categorize the range of possible DCV systems. The generation of pollutants comprised of a constant part, $S_{background}$, associated with the background emission from the building, furniture, etc. and an step-wise constant part associated with the occupants, $S_{occupants}$. The pollutants were assumed to be additive resulting in a step-wise constant emission profile. The pollutant profile was described by the emission ratio (ER) relating the emission during occupied hours to unoccupied hours:

$$\mathrm{ER} = \frac{S_{background} + S_{occupant}}{S_{background}} \tag{4.14}$$

To determine optimum air flow rates for occupancy controlled DCV systems the ventilation effectiveness was calculated for three scenarios using representative values for CAV air flow rate, occupied hours and emission ratios. Systems can have equivalent dose but different cyclic concentration profiles resulting in different peak concentrations. To evaluate the overall air quality performance of the systems the acute-to-chronic exposure represented by peak to average concentration was calculated using equations 4.6 and 4.8. The three analyzed scenarios were:

Scenario 1 evaluated the effect of increasing the ventilation rate when people are present. The generation of pollutants was constant (ER=1) and the reference CAV rate was 0.5 h^{-1}. The number of occupied hours was based on studies of occupancy in buildings that showed that people in general spend 16 hours a day in their home [2, 3]. To cover upper and lower limits of people's presence in their home occupancies of 8 and 20 hours were also analyzed.

Scenario 2 evaluated the effect of increasing the ventilation rate when people are present and more pollutants are emitted during these hours. A reference CAV rate of 0.5 h^{-1} was used and it was assumed people were present in the home 16 hours a day. Emission ratios were deduced from ASHRAE standard 62.2 [51] and EN15251 [45] that both use floor area and number of occupants to specify continuous ventilation requirements. The floor area is related to the emission of pollutants from the building and the number of occupants is related to the additional emission of pollutants due to occupants. In this study it is assumed that pollutant emission rates are proportional to the air flow rates in the standards. The emission ratios for a home of 120 m^2 and varying number of occupants are given in the left graph in figure 4.8. A common occupancy for the home is estimated to be 2 to 3 people, which means that ER equals approximately 1.5. On the right graph in figure 4.8, minimum, maximum and mean emission ratios for homes of 60 m^2, 120 m^2 and 180 m^2 and expected occupancies have been calculated from the two standards. The average value for all three homes is approximately 1.5 and the effectiveness was evaluated using this value.

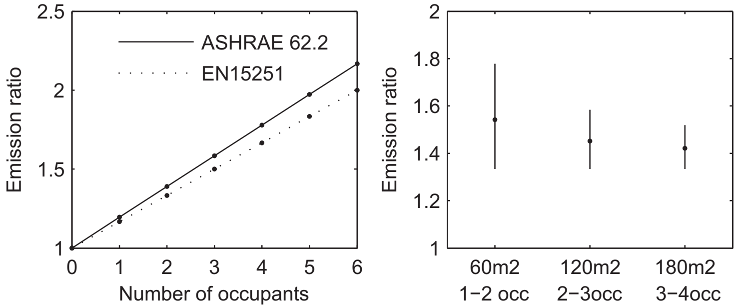

Figure 4.8: Left figure: Emission ratios for a 120 m^2 home with various number of occupants. Right figure: Emission ratio for typical matching home sizes and occupants

To examine the effect of using different values of ER, cases with upper and lower limits of ER equal to 1 and 4 were analyzed. The emission ratio of 1 corresponds to occupants emitting no pollutants. The emission ratio of 4 corresponds to people being the main pollutant source. This high ER case is of increasing interest as occupant generated pollutants becomes more important due to the development, regulation and labeling (e.g., California Environmental Protection Agency composite wood product Airborne Toxic Control Measure [84] and Danish Indoor Climate Labeling [85]) of low emitting buildings materials and furniture.

Scenario 3 evaluated different reference CAV rates. This was done for a case with 16 occupied hours and an emission ratio of 1.5. The CAV air flow rates were selected based on residential ventilation requirements. The ventilation required in residential buildings in Denmark [4] corresponds to 0.5 h^{-1}. The ventilation required by ASHRAE 62.2 is approximately 0.35 h^{-1} including a credit for infiltration and this was used as a lower boundary for the CAV rate. Furthermore, an upper limit for the CAV rate of 1.0 h^{-1} was analyzed.

The effectiveness and acute-to-chronic exposure for the three scenarios are given in figure 4.9 as a function of the low-ventilation factor, LVF. The three graphs of ventilation effectiveness in the first column all show an effectiveness of 1 at the upper boundary of LVF. This was expected as the upper LVF boundary is identical to the CAV system used as the reference case. At the lower boundary of LVF where there is no ventilation during unoccupied hours it can be observed that ventilation is linearly related to the number of occupied hours in scenario 1 (ε equals 1) but not in scenario 2 and 3. E.g. when the home is occupied 8 hours per day the air flow rate needs to be tripled during those 8 hours compared to the reference CAV system to obtain the same dose. Because the source is emitted constantly (ER=1) the CAV system is the case of perfect dilution and the equations derived in section 4.4.1 are all valid. The linearity in scenario 1 is thereby also found when using the equation for intermittent ventilation (eq. 4.13) to calculate

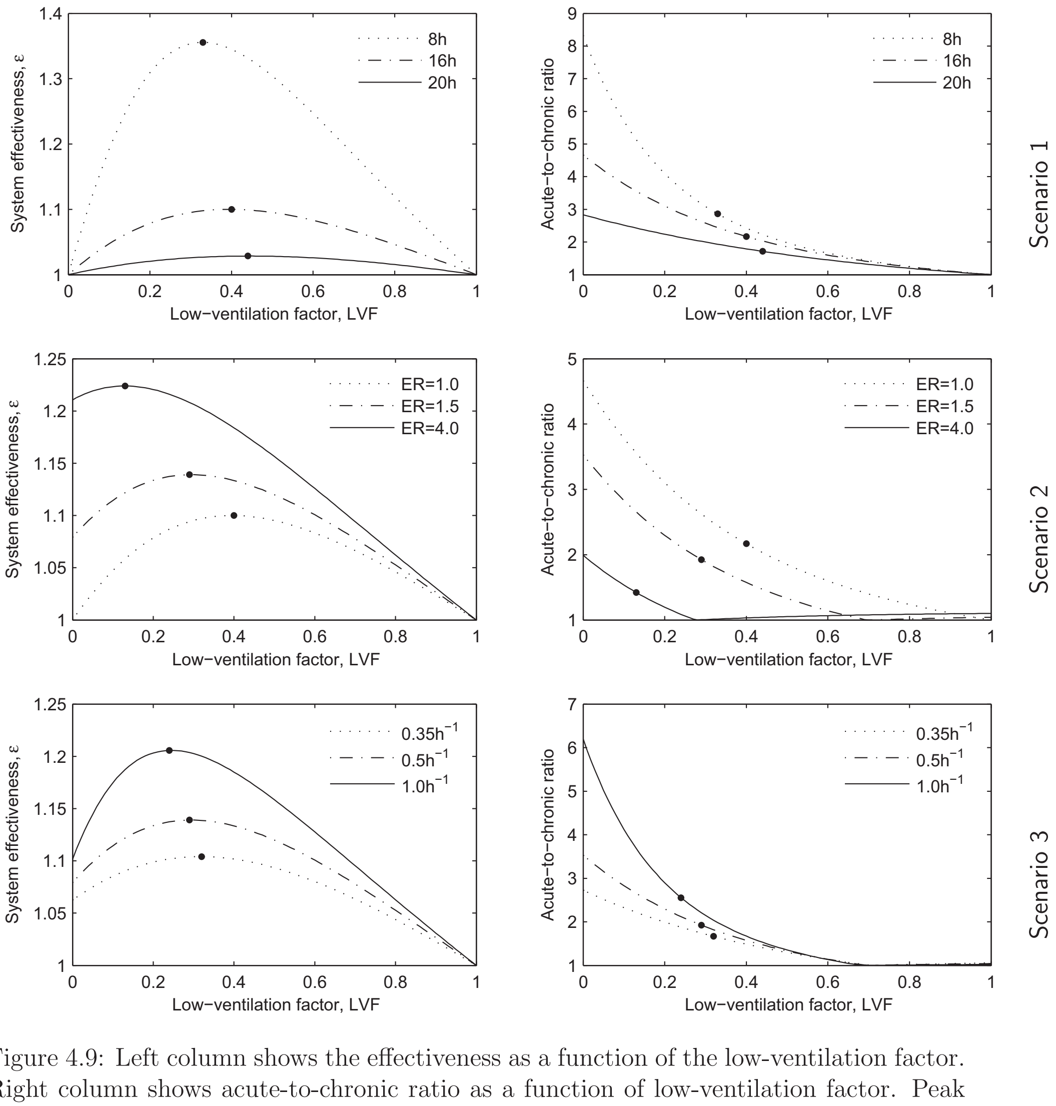

Figure 4.9: Left column shows the effectiveness as a function of the low-ventilation factor. Right column shows acute-to-chronic ratio as a function of low-ventilation factor. Peak effectiveness' are marked by a dot on the curves.

effectiveness ($r = T/t_1$ and $W_1{=}T/t_1 \Rightarrow \phi{=}0$, hence $\varepsilon{=}1$). In scenario 2 and 3 the effectiveness is above 1 at the lower boundary of LVF which means that an intermittent ventilation pattern use less air on a daily basis compared to a constant air flow rate to provide the same dose during occupied hours.

Overall, the results of ventilation effectiveness show that the performance of a DCV system can be optimized given occupancy time and emission ratio. The effectiveness generally increases with fewer occupied hours, higher ER and higher reference CAV rate. Despite the variation of the parameters the three scenarios have many common characteristics. Firstly, all values of peak effectiveness lie within a limited range from 1.03 to 1.36. Furthermore, none of the investigated cases had an effectiveness below 1. This means that one can expect reductions in total volume of exchanged air up to 26% by redistributing the air to times of occupancy and never use more air than in the reference CAV case. The study has thereby demonstrated an upper limit to the theoretically expected reductions under the given assumptions. A reasonable estimate of the expected reduction in total volume of air is 12% representing the case of 16 occupied hours, a reference CAV rate of 0.5 h^{-1} and an emission ratio of 1.5. Another common characteristic is that the low-ventilation factor ranged from 0.13 to 0.44 at peak effectiveness. This means that peak effectiveness occurred when the low air flow rate was 13% to 44% of the reference CAV rate independent of occupancy, emission ratio and reference CAV rate. At peak effectiveness the high air flow rate ranged from 108% to 155% of the reference CAV rate. The highest air flow rate (155% of the CAV rate) occurred in the system with 8 occupied hours, a reference CAV rate of 0.5 h^{-1} and ER=1. All other systems had a high air flow rate in the range 108% to 120% of the CAV rate at peak effectiveness. For the case where the CAV rate is 0.5 h^{-1} and the emission ratio is 1.5 the low and high air flow rates at peak effectiveness are 0.29 and 1.17 respectively. This corresponds to air change rates of approximately 0.15 h^{-1} and 0.6 h^{-1} respectively. The low rate is very similar to the value recommended by EN15251 and given by the Swedish building code of 0.14 h^{-1} during unoccupied hours [45, 46]. The high rate is inevitable higher, because the occupancy based DCV system ensure an average occupant exposure equivalent to the CAV system. By pairing the air flow rates of the examined cases that provided peak effectiveness, the high to low air flow ratio ranged from 2.5 to almost 9. This ratio is of interest when sizing ducts and selecting fans. The largest differences in high to low air flow ratio occurred in the system with 16 occupied hours, a reference CAV rate of 0.5 h^{-1} and ER=4. This change in flow ratio was primarily due to a reduced low air flow rate. All other systems had a high to low air flow ratio of 2.5 to approximately 5 at peak effectiveness.

A consequence associated with dose based design of a DCV system is that the peak concentration changes. At peak effectiveness the highest acute-to-chronic ratio (in this study represented by the peak to average concentration during occupied hours) was below 3 (see dots on the three graphs in the right column of figure 4.9). To determine if peak concentrations are an issue of concern when the chronic exposure (dose) set by the reference case is met, the acute-to-chronic exposure derived from health standards was considered. A literature review of reported chemical pollutants in residences identified 23 pollutants of concern as chronic hazards [83]. The acute-to-chronic ratio for these priority pollutants was determined based on published health standards by dividing the most conservative acute standards by the most conservative chronic standard for lifetime exposure [86]. The health standards base acute exposures on 1, 8 or 24 hour averaged values. This study used peak concentration (an instantaneous value) as the acute exposure value. Averaging of the peak concentration over one or more hours will therefore lead to lower acute-to-chronic ratios. The pollutants with the lowest allowable acute-to-chronic ratios in the three time frames were $PM_{2.5}$, NO_2 and formaldehyde with ratios of 2.5 (24h average), 5.4 (8h average) and 4.7 (1h average) respectively. Because outdoor air can be a significant source of particulates formaldehyde was used as the limiting case. Therefore, if the ratio of the acute-to-chronic exposure in the DCV systems is below 4.7 then the peak concentrations are acceptable. The results showed that the ratio is always less than 3, meaning that the peak concentrations are acceptable and not a barrier to adoption of the DCV technique in residential applications. The results also showed that if occupants contribute to the majority of emissions (e.g. ER=4) then acute-to-chronic ratios may be lower for the DCV system than for a CAV system. In the limit one only needs to ventilate when the home is occupied.

4.5 Discussion

The results in this chapter are based on a range of assumptions e.g. the ventilated space was a perfectly mixed single zone, the load of different pollutants could be added, the hours of occupancy were fixed and consecutive, exposure (and dose) was linearly proportional to the pollutant concentration, pollutants were only removed by ventilation and not by other mechanisms such as filtration or, sorption on surfaces. The results provide an estimate of the expected impact of occupancy based DCV in residential buildings but are due to the assumptions not necessarily applicable outside that range and not definitive in the real world. The single zone and perfectly mixed assumption represents a simplification of occupants' exposure to pollutants. More detailed results can be obtained with multi-zone or CFD simulations. However, such simulations require further assumptions e.g. on the spatial distribution of pollutants and occupants location in the home. Residential DCV studies including spatial distribution of pollutants and the probability of occupancy presence have been made by Laverge [42].

The results of the study showed that it is possible to optimize the demand controlled air flow rates to reduce the quantity of air used for ventilation. For a home occupied 16 hours a day reductions in total volume of exchanged air is about 12%. The trade off is an increased peak concentration. However, the increase in acute-to-chronic exposure is well below the acute-to-chronic exposures of concern derived from health standards. The savings obtained by the theoretical equivalent dose calculations (up to 26%) are relatively small compared to some of the reported savings for residential DCV systems, see table 2.1. However, the occupancy controlled DCV system ensure the same dose during occupied hours as the reference CAV system which is not necessarily the case in those studies that showed higher savings.

The ventilation effectiveness is a first step to evaluate the effect of the occupancy based DCV system on the energy consumption. The analysis did not examine the actual energy implications of the DCV system. Detailed analyses that combine weather data and the time-varying air flow rates of the DCV system are required to evaluate the actual energy performance and is an issue for future work. Energy saving for heating are not necessarily expected to be proportional with ventilation effectiveness as the reduce air flow rates occur at daytime whereas the temperature difference between indoor and outdoor is highest during night time.

Chapter 5

System design

This chapter is primarily based on Paper III where system designs for demand controlled ventilation in multi-family dwellings were analyzed. The paper is summarized in section 5.1. In addition to Paper III, section 5.2 includes an analysis of how the location of the pressure sensor impacts the annual electricity consumption.

5.1 System design for DCV in multi-family dwellings

The objective in Paper III was to develop a simple, inexpensive and energy-efficient centrally balanced mechanical ventilation system for multi-family dwellings. The centralized system was equipped with an efficient heat exchanger and possible savings on heating due to air flow reductions were thereby significantly reduced (see table 2.2). In relation to energy-efficiency, the focus was therefore on the control system and electricity consumption. The following set of performance requirements to the system design was established:

Air flow control

The air flow to each dwelling should be individually controllable.

The control system should ensure balance in air supply and exhaust in each dwelling.

Cost-effectiveness

The initial cost of the system should not exceed 3400 DKK for a dwelling of 70 m^2. This is the net present value of the additional savings by implementing DCV in a ventilation system already equipped with an efficient heat exchanger.

Indoor environment

Air should be supplied to living areas and exhausted from high pollutant rooms (e.g. kitchens and bathrooms).

Draught and dumping of cold air should be avoided.

Diffusers should not produce disturbing noise.

The air flow to a room should not vary more than ±10% on supply flow and ±15% on exhaust flow [87].

The specified performance requirements regarding air flow control allows for two DCV approaches: dwelling-specific flow control or room-specific flow control. Generic system designs are seen in figure 5.1.

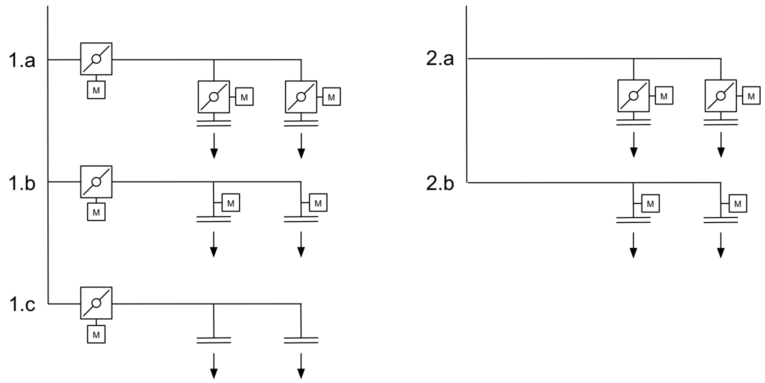

Figure 5.1: Generic system designs for dwelling-specific and room-specific DCV

The simplest layout is system 1.c where a single damper controls the air flow to each dwelling. In this design it is not possible to control the air flow individually for each room. In order to have air flow control for each room it is necessary to apply motorized dampers or motorized diffusers in each room. Systems 1.a, 1.b, 2.a and 2.b show layouts suitable for individual room control. Motorized inlet diffusers (system 1.b and 2.b) make it possible to maintain a constant throw by varying the inlet area. Systems with fixed diffusers will have a variable throw, and must be designed to avoid draught and to ensure reasonable mixing.

Cost-effective system design

The initial cost of a system increases with the number of control components. Furthermore, more control components result in more data signals to manage and maintain and thereby higher cost and complexity. Also, the energy to operate more control components per dwelling can become a substantial part of the total running cost. The initial cost of a room-specific DCV system is expected to exceed the estimated net present value of the savings and will therefore not be cost-effective in multi-family dwellings at the moment. The performance of the dwelling-specific DCV layout (system 1.c) is therefore analyzed further with regard to its control strategy and diffuser types. Furthermore, because occupant density in dwellings is typically lower and less variable than in non-residential buildings, occupancy detection is estimated to be an acceptable an inexpensive control variable for the DCV system. Therefore, the air flow to each dwelling is varied step-wise between fixed air flow rates. A low air flow to dilute constantly emitted pollutants when people are absent denoted 'q_L'. A basic flow to dilute occupant associated pollutants denoted 'q_B' and a forced flow denoted 'q_F' to dilute/remove pollutants associated with

activities such as cooking, showering etc. The low and basic rates are set based on the theoretical calculations of occupancy based DCV systems that comply with a constant air flow requirement of 0.5 h^{-1} with regard to long-term chronic exposures (see section 4.4.2). With a typical occupancy of 16 h per day in homes and a moderate change in pollutant emission during occupied and unoccupied hours the air flow rate can be reduced to 29% of the CAV rate during unoccupied hours and increased to 117% of the CAV rate during occupied hours without introducing problematic short-term acute conditions. The forced rate is set to double the basic rate.
The static pressure reset (SPR) strategy described in section 2.2.3 is applied to the dwelling-specific DCV system. First, control strategies that set the boundaries of the annual electricity consumption are analyzed: a control strategy with fixed static pressure set point sets the upper boundary and a closed control loop with ideal reset of the static pressure set point sets the lower boundary. Then an open control loop that uses predetermined damper positions and a static pressure reset schedule based on the air flow to the dwellings is analyzed.

5.1.1 Analyzed system

A building with 6 dwellings of 70 m^2 was the basis for the analyses. Each dwelling has a bedroom, a living room, a bathroom and a kitchen. The minimum air flow rate to each dwelling is 7 l/s and the maximum flow is 56 l/s. The air flow at basic level is 28 l/s. The program PFS [88] was used to calculate air flows and pressure losses in the circular duct system. The exhaust system is similar to the supply side and operated as a slave and therefore only the supply side is analyzed. A layout of the system is seen in figure 5.2.

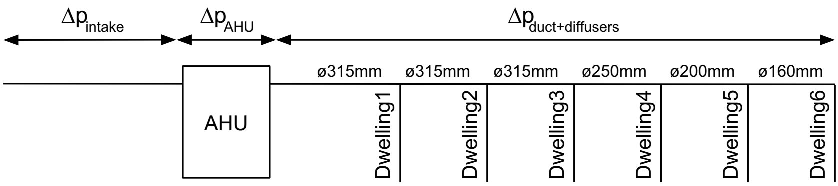

Figure 5.2: Layout of the supply side of the analyzed system including intake, AHU and duct diameters of the distribution system

System curves that show how system losses change with the air flow are made. The curves include all resistances along the flow path from intake to supply diffusers. The pressure loss between two points along the flow path can be expressed by [89]:

$$\Delta p = kq^n \tag{5.1}$$

Where k is a constant that depends on system design, q is the air flow and n is the flow exponent that is 1 for fully developed laminar flow and 2 for fully developed turbulent flows.

Diffusers

The two non-motorized diffusers in figure 5.3 were analyzed: A circular diffuser with a fixed outlet area and a diffuser with self adjusting vanes that change position as the pressure upstream of the diffuser changes and thereby maintain a more stable inlet velocity. The diffuser with variable outlet area is denoted an active diffuser the other a circular diffuser.

Figure 5.3: Investigated non-motorized diffusers. Left figure: Circular air diffuser with fixed opening area. Right figure: Active diffuser with self adjusting inlet vanes that changes the outlet area depending on the pressure upstream of the diffuser

The flow exponent of the circular diffuser is 2 and the flow exponent of the active diffuser is approximately 1.3 according to manufacturer data [90]. The throw of the diffusers was estimated by literature where the throw of a radial jet that flows along a surface is calculated by [52]:

$$x = \frac{Kq}{v_x\sqrt{A_o}} \tag{5.2}$$

Where x is the throw, v_x is the throw velocity usually taken as 0.2 m/s, K is a constant depending on the design of the diffuser, A_0 is the outlet area and q is the air flow. The relative throw, x_r, of a diffuser with a constant outlet area, assuming K remains constant, is calculated by:

$$x_r = \frac{x}{x_{max}} = \frac{q}{q_{max}} \tag{5.3}$$

The throw of a diffuser located in the middle of a room should not exceed half the room width plus the distance from the ceiling to the occupied zone (1.8 m above floor level) to avoid problems with draught. Hence, in a room that is 4 m wide and 2.5 m high, the throw can be 2.7 m. The total air flow to the dwelling is split in two equal amounts to the living and bedroom and each diffuser supplies 3.5 l/s, 14 l/s and 28 l/s at low, basic and forced air flow respectively. It was assumed that the throw of both diffusers was 2.7 m at 28 l/s, hence the relative throw of the circular diffuser is 0.5 at basic flow. This throw is shorter than half the room width and may therefore cause draught. The throw of the active diffuser is expected to vary less due to its variable outlet area and draught issues are therefore expected to be less. The throw at low flow is not relevant regarding draught as residents are absent. The throw of both diffusers at basic flow and their effect on the indoor environment (e.g. noise, dumping of cold air at low supply temperature, mixing of air within the room) should be investigated further to disclose if the performance of the diffusers are acceptable. Regarding energy consumption it was assumed that both diffusers require 20 Pa to supply 14 l/s and equation 5.1 was used to calculate pressure drops at other air flow rates.

Air flow control strategies

System curves for three control strategies and the two diffuser types were generated from a representative selection of the 729 (3^6) possible combinations of q_L, q_B and q_F flows. The investigated control strategies are listed below:

Fixed static pressure This control strategy sets the upper bound for the electricity consumption. The location of the static pressure sensor impacts the power consumption of the fan and the annual electricity consumption was calculated for a system where the sensor was located between dwelling 5 and 6. The static pressure set point was fixed at 92 Pa in the system with the circular diffuser and at 61 Pa in the system with the active diffuser.

Ideal static pressure reset This control strategy sets the lower boundary for the electricity consumption. The static pressure set point is always reset to the minimum level needed to distribute the air which depends on the air flows to the dwellings and their location in the distribution system.

SPR schedule with preset damper positions This control strategy used a schedule in combination with preset damper positions to reset the static pressure set point at part load. A control algorithm collects the air flow requirement of each dwelling and uses the maximum requirement to set the pressure. The issue of occupancy and activity detection is beyond the scope of this paper, but it is assumed to be installed in the homes and incorporated in the control algorithm. The pressure set point can switch between three levels (p_F, p_B and p_L) corresponding to the three possible air flows to a dwelling (q_F, q_B and q_L). The three pressure levels depend on the air flow to each dwelling and on the location of the pressure sensor. In this example the sensor was located in the main duct between dwelling 5 and 6. The highest pressure level, p_F, is therefore identical with the pressure set point in the system with fixed static pressure control that was set high enough to avoid that dwellings are starved of air flow. The dwellings with the highest flow requirement in each unique set of the 729 possible air flow combinations have fully open dampers. Therefore, one damper is always fully open. Dwellings that require air flows lower than the maximum to one of the dwellings have dampers with partly closed positions. At least three positions must be predetermined when the flow to a dwelling can vary between three fixed air flows. The minimum number of positions is only possible when the air flow ratios are identical i.e. q_F/q_B equals q_B/q_L and the positions thereby can be reused in the three pressure levels. The left graph in figure 5.4 illustrates this configuration for a damper with quadratic pressure loss. It is seen that the fully open position is used at each pressure level. Changing the set point scales the flow up or down but the ratio between the flows remains the same.

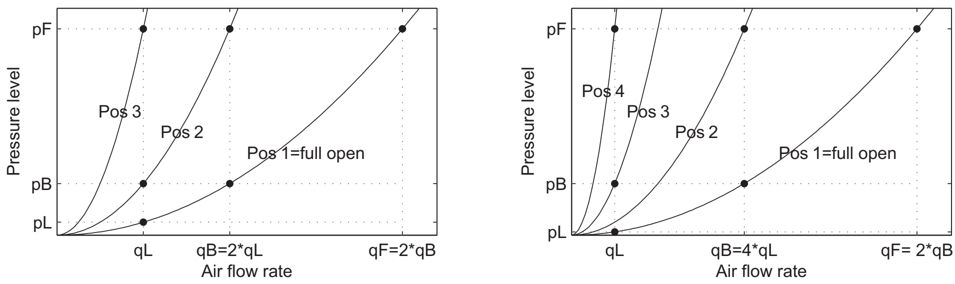

Figure 5.4: Left figure: System with three preset damper positions. Right figure: System with four preset damper positions

The air flow ratios are not identical in the investigated system (q_F/q_B=2 and q_B/q_L=4) as shown in table 5.1. An extra position must be introduced to obtain the possible flow combinations at the three pressure levels, see right graph in figure 5.4. The positions are ranked in ascending order according to the flow ratios. Position 3 that only is used with pressure level p_B is more open than position 4 but more closed than position 2.

Table 5.1: Static pressure set point, air flows, air flow ratios and damper positions in a system with preset damper positions

Static pressure set point (circular / active diffuser)	Air flows	Air flow ratios	Damper position
p_F (92 Pa / 61 Pa)	q_F	$\frac{q_F}{q_F} = 1$	1
	q_B	$\frac{q_F}{q_B} = \frac{56}{28} = 2$	2
	q_L	$\frac{q_F}{q_L} = \frac{56}{28} = 8$	4
p_B (23 Pa / 23 Pa)	q_B	$\frac{q_B}{q_B} = 1$	1
	q_L	$\frac{q_B}{q_L} = \frac{28}{7} = 4$	3
p_L (1.4 Pa / 3.5 Pa)	q_L	$\frac{q_L}{q_L} = 1$	1

Before determining the damper positions the pressure variations occurring when the air flow to the dwellings change needs to be considered. Varying the flow to one dwelling affects the pressure in the entire system and thereby the flow to other dwellings. In systems with motorized control components these instabilities are adjusted for by the control component that continually adapts to the current condition in the system. This is not possible in a system with preset damper positions. Pressure instabilities are instead managed by increasing the pressure drop in the branches making them less sensitive to pressure variations. However, with the disadvantage that fan power consumption increases.

The pressure drop, Δp_{ref}, needed in the branch to avoid instabilities is used to express the pressure drop in the branch when the flow varies by ±10% using equation 5.1 (k is given by: $\frac{\Delta p_{ref}}{{q_{ref}}^n}$):

$$\Delta p_{+10\%} = \frac{\Delta p_{ref}}{{q_{ref}}^n}((1+0.1)q_{ref})^n = \Delta p_{ref}(1+0.1)^n \tag{5.4}$$

$$\Delta p_{-10\%} = \frac{\Delta p_{ref}}{{q_{ref}}^n}((1-0.1)q_{ref})^n = \Delta p_{ref}(1-0.1)^n \tag{5.5}$$

The difference between the two pressure drops calculated in equation 5.4 and equation 5.5 is then used to determine the pressure drop in the branches, Δp_{ref}, when the flow in the duct is predominantly turbulent i.e. n=2:

$$\Delta p_{ref} = \frac{\Delta p_{+10\%} - \Delta p_{10\%}}{(1+0.1)^2 - (1-0.1)^2} = \frac{\Delta p_{+10\%} - \Delta p_{10\%}}{4 \cdot 0.1} \tag{5.6}$$

Equation 5.6 shows that flow variations are estimated to be less than ±10% if the pressure drop in the branches to the dwellings (sum of pressure losses in damper, branch duct and diffuser) is 2.5 times higher than the pressure variations they experience.

This design criterion is applied to the system design. The branches are most sensitive to pressure instabilities when their pressure drop is low, i.e., when dampers are fully open. The maximum recorded pressure variation at a representative number of flow combinations is 8.5 Pa. This means that when dampers are fully open and have forced flow the branch should have a pressure drop of at least 21 Pa to avoid air flow fluctuations more than ±10% of the desired flow. Both diffusers have pressure drops significantly above this level at maximum flow (circular diffuser: $\frac{20\text{ Pa}}{(14\text{ l/s})^2} \cdot (28\text{ l/s})^2$=80 Pa, active diffuser: $\frac{20\text{ Pa}}{(14\text{ l/s})^{1.3}} \cdot (28\text{ l/s})^{1.3}$=49.2 Pa) and flow fluctuation due to instability are not expected to be a problem. The four positions at each damper are determined by applying the static pressure reset schedule to a system with free dampers. A flow calculation with the preset positions and the SPR schedule showed that all flow deviations were within ±10%.

The systems annual electricity consumption was calculated using the generated system curves, a diurnal occupancy schedule and fan efficiency curves. The annual electricity used to transport the air is seen in figure 5.5 assuming identical supply and exhaust systems. The average SFP value of the systems is given on the bars.

The system with the preset damper positions and SPR schedule has slightly higher electricity consumption than the ideal SPR strategy that sets the lower boundary for the electricity consumption. The electricity consumption is approximately reduced by 30% in the systems with circular diffusers and 20% in the systems with active diffusers when resetting the pressure set point at part load conditions compared to systems with the same diffusers and a fixed static pressure control.

The two investigated diffusers result in a difference in electricity consumption of approximately 7% when the SPR is employed. The difference is larger with fixed static

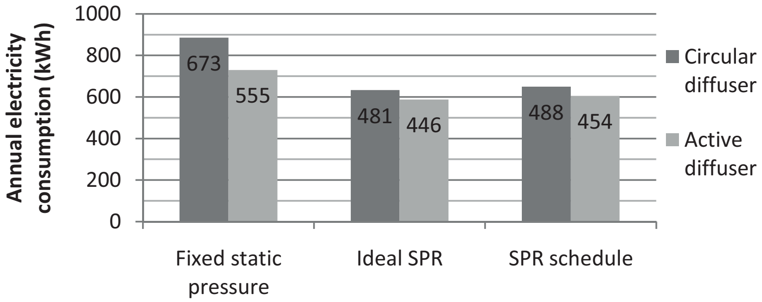

Figure 5.5: Annual electricity consumption for ventilation of the 6 dwellings. The systems average SFP value flow are given on the bars.

pressure control (approximately 17%). The preferred diffuser should be the one that provides best energy and indoor environmental performance. The throw of the diffusers at basic flow and the impact on the indoor environment should be investigated in future work to determine the best performance. The active diffuser that changes the outlet area depending on the upstream pressure is an interesting component for variable air flow ventilation in dwellings regarding its throw and potential fewer draught issues.

5.2 Location of pressure sensor

The location of the static pressure sensor impacts the power consumption of the fan and thereby the annual electricity consumption. The annual electricity consumption was calculated for 6 different sensor locations in the main duct: between the AHU and dwelling 1 (labelled ps@AHU) and at the 5 locations between neighboring dwellings (labelled ps@1-2 when the pressure sensor is located in the main duct between dwelling 1 and 2, etc.), see figure 5.2. The closer the sensor is located to the fan the higher set point is needed to avoid dwellings being starved of air. Figure 5.6 show the static pressure set point needed to ensure the designed air flow rates to all dwellings at 6 locations in the main duct.

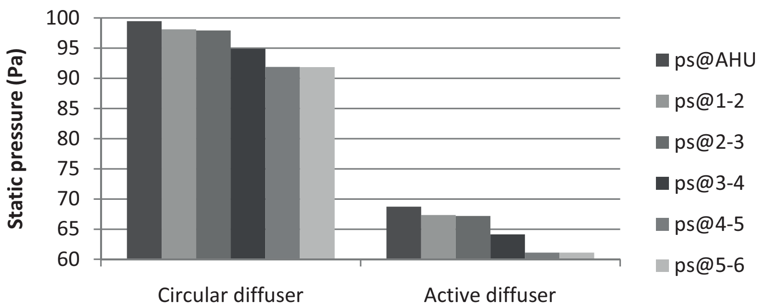

Figure 5.6: Static pressure set point at 6 locations in the main duct for the circular and active diffuser

The annual electricity consumption was calculated for the 6 sensor locations and the two diffusers using the same occupancy schedule as in Paper III. Figure 5.7 shows the impact of the location of the sensor on the annual electricity consumption. The electricity consumption is lowest when the sensor is located between the dwelling 4 and 5. This location reduces the annual electricity consumption by approximately 3 kWh per dwelling compared to a sensor location close to the fan and independent of the diffuser type.

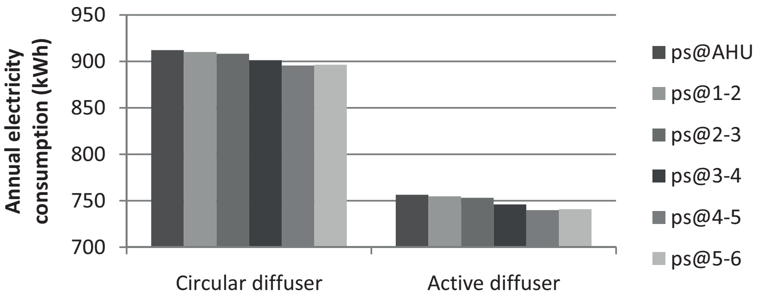

Figure 5.7: Annual electricity consumption for ventilation of the 6 dwellings with 6 different sensor locations and two types of diffusers

5.3 Discussion

The advantage of the SPR strategy is that it saves energy at part load conditions compared to a system with fixed static pressure control. The estimated savings are in the lower range of the saving reported in literature of 30% to 50% [54]. The electricity consumption of the system with fixed static pressure control corresponds to approximately 2.0 kWh/m^2 per year and approximately 1.5 kWh/m^2 when a SPR control strategy is applied. The electricity used to transport the air in a system with fixed static pressure control thereby accounts for approximately 16% of the total energy consumption in a low-energy dwelling class 2015 of 6 x 70 m^2 when the electricity is weighted by a site-to-source factor of 2.5 (see section 2.2.2) whereas the system with a SPR control strategy accounts for approximately 11%. In a low-energy dwelling class 2020 the fixed static pressure control and SPR control account for 19% and 25% of the total energy consumption respectively.

The most favorable location of the pressure sensor was in between the second most remote dwellings (dwelling 4 and 5). This is in agreement with the guidance in ASHRAE Handbook Applications [54] that recommends to locate the sensor 75% to 100% of the distance from the first to the most remote diffuser. However, the savings are very small (approximately 3 kWh per dwelling per year) compared to locations closer to the fan for the investigated system.

The system with preset damper positions requires accurate setting of the static pressure to deliver the desired flows. Even small deviations from the required pressure can result in considerable flow changes. Poor air flow precision is undesirable and should not be encouraged at the expense of reducing energy consumption. The lowest pressure level is only used when all apartments are unoccupied and the situation will probably seldom occur. Furthermore, the level will most likely be unattainable to hold due to among others the influence of natural driving forces as mentioned by Schild [91]. A tradeoff can be to design the control algorithm to maintain a higher pressure and not have fully open dampers at this combination of air flows.

Balancing and commissioning of the developed system can be time-consuming because of the fixed damper positions that need to be tuned. It is often advised to minimize the need for manual balancing of the system by using automatically adjusting control components [92]. However, systems with motorized control components also need to be tuned to obtain optimal performance. Rational procedures to balance/tune a system with preset damper positions should be examined in future work. Implementation of the SPR schedule with preset damper positions requires dampers that easily can switch between the four predetermined positions after receiving a signal from the control algorithm. A damper fulfilling this criterion should be developed. If the cost of motorized control components decreases the reset schedule could use these to switch between the three flows.

Chapter 6

Conclusion

This project dealt with demand specification and system design of demand controlled ventilation for multi-family residential buildings. A literature study of pollutants in homes, their sources and their impact on humans formed the basis for the demand specification. Theoretical analyses of demand specification focused on the time-varying air flows needed to provide an average occupant exposure equivalent to that obtained when meeting current codes and standards requirements for constant air flow. The analyses showed that:

- The time-varying air flow rates of an occupancy based DCV system can reduce the average concentration of background and occupancy related sources and the peak concentration of the occupancy related source without exchanging more air than a system with constant air flow. However, this is at the expense of increasing the peak concentration of the constantly emitted background source.
- Pollutants emitted constantly by background sources such as building materials, furniture etc. are the limiting factor for DCV systems to provide a peak occupant exposure equivalent to that in a system with constant air flow.
- Time-varying air flow rates based on occupancy can reduce the volume of exchanged air by up to 26% compared to a system with constant air flow while maintaining the same average occupant exposure.
- The greatest reductions in exchanged volumes of air were obtained when the air flow during unoccupied hours was 13% to 44% of the CAV rate and the air flow during occupied hours was 108% to 155% of the CAV rate.
- The volume of exchanged air can be reduced without introducing problematic acute conditions.
- The results of the project provide more flexible approaches to ventilation design for residences that allow occupancy based DCV approaches to comply with codes and standards that are currently based on continuous ventilation rates.

The part of the work dealing with system design focused on simple and cost-effective centrally balanced DCV systems for multi-family dwellings. A design expected to meet this requirement was investigated in detail with regard to its electricity consumption by evaluation of different air flow control strategies. The analyses of potential system designs showed that:

- The initial cost of the system should not exceed 3400 DKK for a dwelling of 70 m^2. This is the net present value of the additional savings by implementing DCV in a ventilation system already equipped with an efficient heat exchanger.
- A system design where a single damper controls the air flow to each dwelling is expected to be a cost-effective and simple solution.
- A control strategy that resets the static pressure set point at part load and uses preset damper positions reduces the electricity consumption by 20% to 30% compared to a system controlled by constant static pressure.
- The pressure sensor should be located in the main duct near the most remote diffuser following current ASHRAE guidance.

6.1 Suggestions to future work

The following suggestions to future work are made:

- Detailed analyses that combine weather data and the time-varying air flow rates of the DCV system are needed to evaluate the actual energy performance of the system.
- Analyses that combine the single zone and perfectly mixed assumptions used in the equivalent dose approach in section 4.4 with spatial ventilation efficiency will provide more detailed information with regard to occupant exposure.
- Analyses that specify how uncertainties in emission ratio, reference CAV rate and the number of occupied hours impact the ventilation effectiveness can be used to predict uncertainties in the performance of the system.
- Analyses that investigate the effect of the non-motorized diffusers on the indoor environment (e.g. noise, dumping of cold air at low supply temperature, mixing of air within the room) should be carried out to disclose if the performance of the diffusers are acceptable.
- Experiments that investigate the performance of a system with preset damper positions regarding air flow precision, pressure instabilities etc. are needed to determine possible application of the system.
- Development of dampers that switch between fixed positions and rational procedures to balance/tune a system with preset damper positions are needed to apply the system.

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Part II

Appended papers

Paper I

'Derivation of Equivalent Continuous Dilution for Cyclic Unsteady Driving Forces'

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Derivation of equivalent continuous dilution for cyclic, unsteady driving forces

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ABSTRACT

This article uses an analytical approach to determine the dilution of an unsteadily-generated solute in an unsteady solvent stream, under cyclic temporal boundary conditions. The goal is to find a simplified way of showing equivalence of such a process to a reference case where equivalent dilution is defined as a weighted average concentration. This derivation has direct applications to the ventilation of indoor spaces where indoor air quality and energy consumption cannot in general be simultaneously optimized. By solving the equation we can specify how much air we need to use in one ventilation pattern compared to another to obtain same indoor air quality. Because energy consumption is related to the amount of air exchanged by a ventilation system, the equation can be used as a first step to evaluate different ventilation patterns effect on the energy consumption. The use of the derived equation is demonstrated by representative cases of interest in both residential and non-residential buildings.

1. Introduction

The issue being addressed in this article is how to treat a cyclic process in which a variably generated solute is being diluted by a variable flow-rate solvent in a stream. The goal is to find a simplified way of showing equivalence of such a process to a steady-state one.

1.1. Target application

The motivation for this work is to find ways of showing equivalence in determining indoor air quality in buildings when both sources and ventilation rates vary over time. A key step in designing a building is determining the correct amount of ventilation and the optimal system with which to provide it. There is no shortage of guidance on how much ventilation to use. The standard of care for ventilation system design in the US is probably the 62 series of ASHRAE standards (62.1-2004 for non-residential buildings [1] and 62.2-2010 for residential buildings [2]). In Europe the standard EN15251-2007 [3] gives recommended ventilation rates for non-residential buildings but countries specify design ventilation rates in their national codes.

When ventilation rates are stated in terms of airflow rate per person (e.g. l/s pr. person) or airflow rate per floor area (e.g. l/s pr. m^2), we generally assume a constant airflow during the entire period of interest. There are, however, a variety of reasons why one might want to design and operate the ventilation system with variable amounts of ventilation airflow. For example:

- There may be periods of the day when the outdoor air quality is poor and one wishes to reduce the amount of outdoor air entering the building;
- Equipment operating for other reasons (e.g. economizer operation) can provide exogenous ventilation from the point of view of indoor air quality and energy savings can be achieved by lowering the designed mechanical ventilation to account for it;
- Energy or power costs may make it advantageous to reduce ventilation for certain periods of the day;
- Some HVAC equipment may make cyclic operation more attractive than steady-state operation such as residential or small commercial systems that tie ventilation to heating and cooling system operation;
- The generation of pollutants indoors may vary over time, e.g., depending on occupancy such that adjusting the ventilation rate according to the demand can improve indoor air quality and potentially save energy;
- Different times of the day may be more important than others (e.g. because different number of occupants are present) and can be weighted differently.

Regardless of the reason, the designer or decision-maker needs a method to determine how two ventilation systems compare for the purposes of providing acceptable indoor air quality. ASHRAE Standard 62.1-2004 does not directly address these issues; 62.2-2010 does address intermittent ventilation compared to continuously operated ventilation system in a limited way. The standard

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Nomenclature

A	dilution rate
C	concentration
d	dose
f	fractional time in the period T
N	dilution
r	fractional dilution rate in the step
S	solute source strength
t	time
T	period
V	volume
W	weighting
Z	simplifying variable

Greek symbol

ε	ventilation effectiveness
ρ	density
Φ	simplifying variable

Subscript

*	reference case

Abbreviations

ACH	air change rate
VOC	volatile organic compounds

EN15665 [4] sets out criteria to assess the performance of residential ventilation systems concerning hygiene and indoor air quality.

1.2. Ventilation background

Ventilation is principally used to maintain acceptable indoor air quality by controlling indoor contaminant concentrations and minimizing occupant exposures to the contaminants. Whole-building ventilation dilutes contaminants in the indoor air with air that does not contain those contaminants, and is normally used for controlling unavoidable, generic or non-specific contaminants. When specific contaminant sources can be identified, they are best dealt with directly through source control methods including local exhaust. For example, bathroom and cooking contaminants (including water vapor) are best addressed by exhaust fans in those spaces. Volatile organic compounds (VOCs) are often best addressed by changes in composition or use of specific materials.

If ventilation rate and contaminant concentration were linearly related, the average concentration would be proportional to the average ventilation and straightforward methods could be used to determine the effectiveness of a ventilation system with variable flow rates. Unfortunately, ventilation and concentration are dynamically and inversely related through the mass continuity equation, which leads to a typically non-linear relationship between ventilation and concentration.

Solutions to the continuity equation always involve an air change rate (ACH)[1] appropriate to the problem at hand. Although we are often more accustomed to dealing with ventilation in terms of specific airflow rates, the efficacy of a ventilation system with variable air flow rates will depend on the air change rate, so it is important to keep typical rates in mind for some specific, but common occupancies. A related parameter of interest is the *turn-over time*, which is the inverse of the air change rate. It is the characteristic time in which the concentration of a contaminant responds to a change in ventilation rate.

One can derive typical air change rates and turn-over times from literature and from standards using specific ventilation rates, typical occupant densities, and typical geometry of the space in question.

The turn-over times in Table 1 vary from 6 min to 6 h, indicating that different occupancies will behave quite differently at a variety of configurations. The use of such quantities to explore the spatial dependency of ventilation is also important for large spaces, but will not be discussed here. Sandberg and Sjoberg [9] developed much of the nomenclature used in this field to deal principally with spatial variation.

[1] The term "air change rate" does not discriminate between, infiltration, natural ventilation, or mechanical ventilation. It simply denotes that air is being exchanged between indoors and outdoors.

Sherman and Wilson [10] followed by Yuill [11,12] have already solved the continuity equation for the general case and defined (temporal) ventilation effectiveness,[2] ε, as a measure of how good a given, time-varying, ventilation pattern is at providing acceptable IAQ. As in those cases, we limit our analysis to contaminants with a linear dose-response and no other loss mechanism (e.g. sorption or deposition). ASHRAE Standard 136 [13] uses this kind of approach to convert time-varying envelope air leakage into an effective seasonal ventilation rate. Sherman [14] studied the case of equivalent dilution of a steadily generated source for intermittent ventilation compared to constant ventilation. These results have been included in ASHRAE standard 62.2 by allowing intermittent ventilation provided that the ventilation rate is raised outside the off period. This paper expands on the work by Sherman to also consider variable source generation, variable flow rates and variable weighting of the concentration. The purpose of this paper is to develop approaches for determining the indoor air quality equivalency of different ventilation systems based on fundamental principles of mass balance. The approaches are demonstrated using a few representative cases of interest in both residential and non-residential buildings.

2. Problem definition

Consider the situation in which we have a small amount of solute being generated at a known time-varying rate $\dot{m}_{solute}$ inside a solvent filled space of volume V. This space is being flushed by the solvent at a known, time-varying rate $\dot{m}_{solvent}$ to yield a time varying solute concentration, C. The relationship between these quantities is constrained by the conservation of mass as follows:

$$V\dot{C} + \frac{\dot{m}_{solvent}}{\rho_{solvent}}C = \frac{\dot{m}_{solute}}{\rho_{solute}} \tag{1}$$

Assuming constant densities for solvent and solute Eq. (1) can be expressed by:

$$\dot{C} + AC = S \tag{2}$$

where $A \equiv \dot{m}_{solvent}/V\rho_{solvent}$ is the dilution rate of the solvent stream and $S \equiv \dot{m}_{solute}/V\rho_{solute}$ is the source strength of the solute.

We are investigating the problem of a system that is unsteady, but in cyclic equilibrium over some known period, T. This means that the concentration, the solute source strength and the dilution rate take on same value when the time changes one period: $C(t-T) = C(t)$, $S(t-T) = S(t)$ and $A(t-T) = A(t)$.

We seek to evaluate some test system that performs dilution equivalent to a reference case. We define the dose, d, as the quan-

[2] We will also use the term *efficacy* as a synonym for (temporal) ventilation effectiveness.

Table 1
Example air change rates and turn-over times

ACH (1/h)	Turn-over time (h)	Description
0.15	6.67	Assumed infiltration rate of homes [2]
0.25	4.00	Infiltration rate of commercial buildings [5]
0.3	3.33	Ventilation requirement of almost empty commercial buildings [1]; estimated infiltration rate of new homes [6]
0.45	2.22	Ventilation requirement for small a homes of 90 m^2 including default 62.2 infiltration credit [2]
0.5	2.00	Office space requirement[1]; also large home [2]; requirements in residential buildings in Denmark [7]
1	1	Infiltration rate of older home [8]
2	0.50	Conference room requirement [1]
4	0.25	High density space (e.g. theater lobby) [1]
6	0.17	School – Low polluted building, Indoor environment category B [3]
10	0.10	School – Non-low polluted building, Indoor environment category A [3]

tity that we wish to hold constant in determining equivalent dilution and the dose is calculated as the weight-integrated concentration over the cyclic period, T:

$$d = \oint C(t)W(t)dt \tag{3}$$

where W is the weighting function: $\oint W(t)dt = T$. The weighting function allows us to emphasize parts of the cyclic period heavier than others or to omit parts of the cyclic period by using a zero value weighting factor when solving for equivalent dose.

2.1. Reference case

Before proceeding further, we consider the reference case to compare our test case with. We select as our reference the case conventionally called *perfect dilution*, which we define as that time varying reference dilution rate $A_*(t)$ that holds the concentration constant at some steady state value, C_*. By inspection of Eq. (2) the time-varying reference dilution rate is:

$$A_*(t) \equiv \frac{S(t)}{C_*} \tag{4}$$

In our reference case the dose is then:

$$d = \oint C_* W(t)dt = C_* T \tag{5}$$

Let us now define the *efficacy* as the ratio of the amount of solvent required in the reference case compared the amount in the test case under consideration. The amount of solvent used in the reference case, N_*, is the integrated reference dilution rate over the cyclic period:

$$N_* \equiv \oint A_*(t)dt \tag{6}$$

The amount of solvent used in the test dilution system, N, is:

$$N \equiv \oint A(t)dt \tag{7}$$

From Eqs. (6) and (7) we calculate the efficacy:

$$\varepsilon \equiv \frac{\oint A_*(t)dt}{\oint A(t)dt} = \frac{N_*}{N} \tag{8}$$

The efficacy is a measure of how much solvent we need to use in our test system compared to the reference case to obtain the same dose. The efficacy can be used as a target or optimization parameter in the design process. We can design our test system to match some target efficacy that often will be unity, as we then provide dilution equivalent to our reference case. The efficacy can also be used as an optimization parameter in designing systems that provide same dose but use less solvent.

3. Derivation of dose for cyclic, unsteady driving forces

We want to derive an equation for equivalent dilution in our test and reference case and we use dose as the quantity that we wish to hold constant in determining equivalent dilution. To evaluate the dose (Eq. (3)) in our test case, we need to solve the continuity equation (Eq. (2)) for the concentration, $C(t)$. The standard integral form of an inhomogeneous, first-order, linear differential equation with arbitrary coefficients can be used to do this:

$$C(t) = C(t_0)\xi(t, t_0) + \int_{t_0}^{t} S(t')\xi(t, t')dt' \tag{9}$$

where $C(t_0)$ is the known constant of integration representing the concentration at some reference time, t_0. For simplicity we have defined the following function:

$$\xi(t, t') \equiv e^{-\int_{t'}^{t} A(u)du} \tag{10}$$

Our process is cyclic over the period T and the concentration therefore takes on the same value when the time changes one period. This also means that t_0 in our constant of integration is arbitrary and the constant of integration itself is a solution to the differential equation. The time-varying concentration in our cyclic process is then given by:

$$C(t) = \frac{\int_{t-T}^{t} S(t')\xi(t, t')dt'}{(1 - \xi(T, 0))} \tag{11}$$

The derivation of the time-varying concentration can be found online in the paper's Supplementary data in Section 1.

By substituting the time varying concentration (Eq. (11)) into the expression of the dose (Eq. (3)) we can calculate the dose for any test system with variable dilution rates, solute source strength and weightings by the following double integral:

$$d = \frac{\oint W(t) \int_{t-T}^{t} S(t')\xi(t, t')dt'dt}{(1 - \xi(T, 0))} \tag{12}$$

With the derived expression of the dose (Eq. (12)) for an unsteady but cyclic test system we can compare this to the dose in our reference case of perfect dilution (Eq. (5)). Depending on which parameters are known we can use the equivalency equation in different ways.

If we know the solute source strength, $S(t)$, and the time-varying dilution rate in our test system, $A(t)$, is completely specified we can determine what the steady-state concentration, C_*, in our reference case of perfect dilution would be. Sometimes we are trying to design a system that produces a dose equal to that in our reference case and we can use Eq. (12) as the constraint on the test system that makes that true. Our problem then reduces to finding that test dilution pattern that gives us the target dose.

Because we defined our time varying reference dilution rate $A_*(t)$ as that which holds the concentration constant at some steady state value, C_*, see Eq. (4), we do not need to individually know the solute source strength, $S(t)$, and the steady state concentration, C_*, but only the presumed dilution for perfect dilution, $A_*(t)$.

4. Step function

The applications we consider further on will only involve situations in which the weightings, solute source strength and dilution rates are all step-wise constant with one step at time t, see Fig. 1. Any of the three parameters can change at the step t_1 or they must remain the same through the cyclic period, T. In other words, they have to change at the same time or not change at all.

Because of the step-wise constant profiles we can expand the dose equation (Eq. (12)) into a sum of integrals where the parameters are constant. We can thereby set up an analytical expression for equivalent dose in our reference and test case (Eq. (5) equals Eq. (12)). This analytical expression can be simplified using the following definitions: weighting of the two periods: $W_1t_1 + W_2(T - t_1) = T$, reference dilution: $N_* = A_{*1}t_1 + A_{*2}(T - t_1)$, test dilution: $N = A_1t_1 + A_2(T - t_1)$, fraction of time in the step: $f \equiv t_1/T$, non-dimensional test dilution rate in the step: $r \equiv A_1T/N$, and a non-dimensional reference dilution rate in the step: $r_* \equiv A_{*1}T/N_*$. The non-dimensionalized dilution rates, r and r_*, will be 1 when the dilution rate does not change through the cyclic period and they will be 0 when the dilution rate during one of the two periods is zero. The parameters r, r_* and W_1 can take on values in the interval $[0, T/t_1]$.

By introducing the variables: $Z \equiv fr$ and $\varphi \equiv f^2[(r - r_*)(r - W_1)]$ the equation for equivalent dilution can be reduced to:

$$N_* = \frac{N}{1 + \frac{\phi}{Z(1-Z)} - \frac{2}{N}\frac{\phi}{(Z(1-Z))^2}\Big/\left(\mathrm{Coth}\left[\frac{NZ}{2}\right] + \mathrm{Coth}\left[\frac{N(1-Z)}{2}\right]\right)} \tag{13}$$

For detailed derivation see the online Supplementary data in Section 2.

Φ and Z are only introduced to simplify the equation and they also allow us to more easily investigate the behavior of our step wise constant problem in its space of solutions. In Eq. (13) it is worth noting that Z is symmetrical around ½ as replacing Z by $(1 - Z)$ yields the same result.

We can derive a recursive expression for the efficacy, but at the expense of breaking the symmetries of Z:

$$\varepsilon = \frac{1 - \left(\frac{\phi}{(1-Z)} - Z(1/\varepsilon - 1)\right)\left(\frac{N_*}{2}\right)\left(\mathrm{Coth}\left[\frac{N_*Z}{2\varepsilon}\right] + \mathrm{Coth}\left[\frac{N_*(1-Z)}{2\varepsilon}\right] - \frac{2\varepsilon}{N_*Z}\right)}{1 - \frac{\phi}{(1-Z)^2}} \tag{14}$$

Because we consider a step-wise constant system we only need to know the ratio of the reference dilution rates (or solute source strength) in the two periods to estimate the efficacy of the system. However, if we only know the ratio we cannot calculate the actual dose.

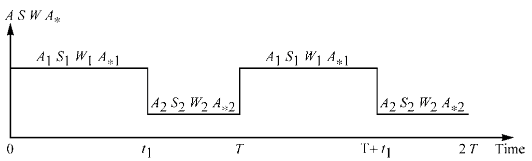

Fig. 1. Step wise constant weighting, solute source strength and dilution rate during the cyclic period, T.

5. Discussion

5.1. Phase space of Φ and Z

Before discussing the actual phase space of the efficacy it is important to realize that the allowable phase space of the parameters Φ and Z is limited. The maximum value Φ occurs when the product: $(r - r_*)(r - W_1)$ is as large as possible. This occurs at two points; the first is when r_* and W_1 equals 0, hence $\Phi_{max} = Z^2$. Because Z is symmetrical around ½, $\Phi_{max} = (1 - Z)^2$ for $Z > ½$. The minimum value of Φ occurs when one of the differences: $(r - r_*)$ or $(r - W_1)$ is positive and the other is negative. Because r_* and W_1 take on values in the interval $[0; 1/f]$, $\Phi_{min} = -Z(1 - Z)$. The limits of Φ are thereby given by: $-Z(1 - Z) \leqslant \phi \leqslant Maximum(Z^2, (1 - Z)^2)$. Fig. 2 shows the allowed phase space of Φ and Z. It is seen that Φ goes from zero to unity when $Z = 0$ (or $Z = 1$), but when $Z = ½$, Φ goes between ±0.25.

5.2. Intermittent dilution

In the limiting situation where there is no dilution during one of the two steps Z equals either 0 or 1. We call this limit *Intermittent Dilution*. If we take the limit of Eq. (14) when Z approaches zero (or unity) we get the following expression for the efficacy:

$$\varepsilon_0 = \frac{1}{1 - \phi + \phi(N/2)\mathrm{Coth}\left[\frac{N}{2}\right]} \tag{15}$$

where we have used the subscript on the efficacy to show that it is for a solution where one of the two steps has no dilution. Sherman [14] solved the case of intermittent dilution when the solute source strength and weighting were constant during the cyclic period. In that specific case of intermittent dilution Φ approaches the following limit:

$$\phi \to f^2 \tag{16}$$

For that application the solution is more conveniently expressed as a recursive relationship between the efficacy and the reference dilution, N_*. We can also express the more general intermittent dilution solution (Eq. (15)) in that form as follows:

$$\varepsilon_0 = \frac{1 - \phi(N_*/2)\mathrm{Coth}\left[\frac{N_*}{2\varepsilon_0}\right]}{1 - \phi} \tag{17}$$

5.3. Phase space of efficacy

Let us know examine the phase space of the efficacy by the equation for equivalent dose (Eq. (13)). Fig. 3 is a plot of the

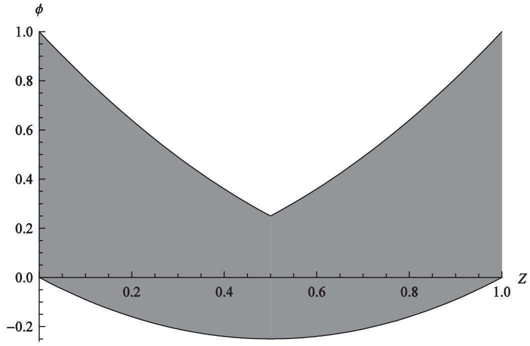

Fig. 2. Allowable phase space of Φ vs. Z.

efficacy vs. reference dilution rate at a representative value of $\Phi = 0.25$. A mesh of curves for different values of Z is graphed spanning the full range of Z. The lower bounding curve is for $Z = 0$ (or $Z = 1$ corresponding to the intermittent dilution limit) and the upper curve is for $Z = ½$.

For reference dilution values, below approximately 2, the efficacy is independent of Z and the solution for intermittent dilution provides sufficiently accurate results. As the reference dilution gets higher the efficacy approaches an asymptote that is very much dependent on Z. Taking the limit of the general equivalency equation (Eq. (14)) for high dilution rates we find that efficacy asymptotically approaches a limit given by:

$$\varepsilon_\infty = \frac{1}{1 + \frac{\phi}{(Z(1-Z))}} \tag{18}$$

This suggests that for practical problems one may choose to use the efficacy equation for intermittent dilution until the efficacy approaches the limit given by the asymptote. Inspection of Eq. (18) also shows us that because Z always is positive, the efficacy is below unity for positive values of Φ and above unity for negative values of Φ. In Fig. 3 we saw how the efficacy depended on Z and not Φ which was maintained at a fixed value of 0.25. Fig. 4 shows the efficacy at three different values of Φ each spanning their individual range of Z.

Again we see that for a test dilution of less than about 2, the intermittent dilution equation (Eq. (15)) provides sufficiently accurate results as Z in this range has little effect on the efficacy. Furthermore we also see how the efficacy can take on values above unity when Φ is negative. At low dilution the efficacy for $\Phi = -0.1$ starts near unity but it can slowly grow without bound for increasing dilution. Efficacies above unity means that a test case can perform better (i.e. use less solvent) than our reference case of perfect dilution. For Φ to be negative, r must be between the values of r_* and W_1. This could occur, for example, in a case where the source was high during a period when the weighting was low. The most negative Φ occurs when the sum of the two difference $(r - r_*)$ and $(r - W_1)$ is a big as possible. This happens when r takes on a value exactly between r_* and W_1, hence: $r = (r_* + W_1)/2$ which means that: $\varphi_{low} = -(f^2/4)(r_* - W_1)^2$.

Fig. 5 shows the dependence on Φ a bit more clearly for representative values of test dilution ($N = 2$, 5 and 10) and spanning the full range of Z. The lower bounding curve is for $Z = 0$ when Φ is positive whereas Z for the upper bounding curve changes depending on Φ. When Φ is negative the lower bounding curve is for $Z = ½$ and Z changes for the upper bounding curve depending on Φ.

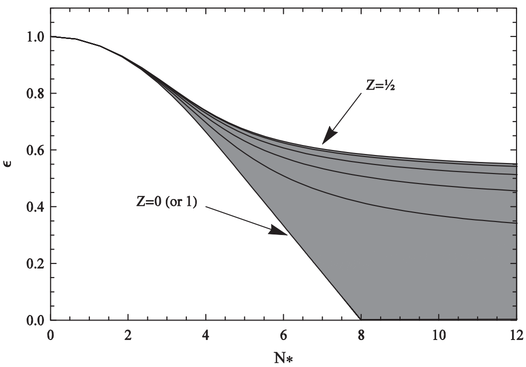

Fig. 3. Efficacy vs. reference dilution for $\Phi = ½$.

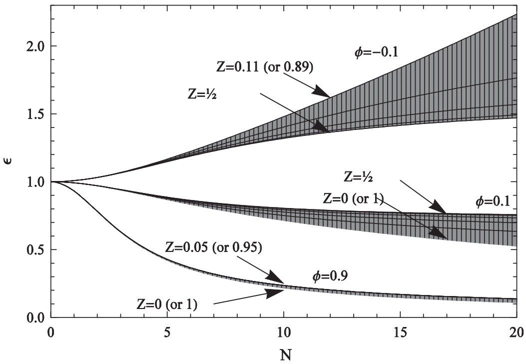

Fig. 4. Efficacy vs. test Dilution for $\Phi = -0.1, 0.1, 0.9$.

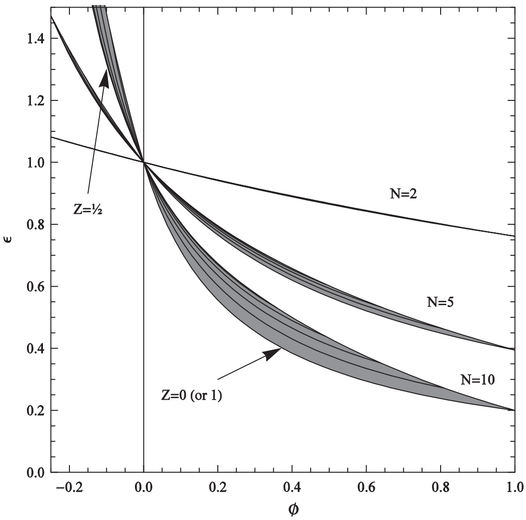

Fig. 5. Efficacy vs. Φ for three different test dilutions ($N = 2, 5, 10$ from flattest to steepest). Mesh is for full range of allowable values of Z.

In Fig. 5 we again see that for low values of test dilution there is not much dependence on Φ for the efficacy but at higher values there is (as is there on Z). Again our results show that for a test dilution of two or below the efficacy is independent of Z and the equation for intermittent ventilation (Eq. (15)) will provide sufficiently accurate results. From Fig. 5 we also see that to design systems with high efficacies the strategy would be to minimize Φ as much as possible. If more degrees of freedom are available, Z can be optimized after that; an optimal Z should be as close to ½ as possible for efficacies below unity and as low (or high) as possible for efficacies above unity.

5.4. Approximate solution

For some applications it may be desirable to have an approximate solution. We note two results from above that suggest an approximate solution. The first result is that high test dilutions solutions are generally low efficacy and relatively independent of

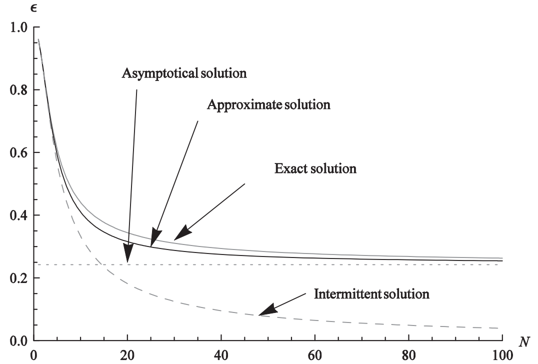

Fig. 6. Approximate, exact, intermittent and asymptotical solution of ε for $\Phi = 0.5$ and $Z = 0.2$.

efficacy (see Fig. 3). The second result is that low test dilution solutions have efficacies near unity and are relatively independent of Z (see Figs. 3–5), and thus equal to the intermittent dilution solution. Accordingly, an approximate solution for the efficacy that works over a broad spectrum combines the limiting solutions is as follows:

$$\varepsilon \approx \varepsilon_0 + \varepsilon_\infty (1 - \varepsilon_0)^2 \tag{19}$$

This approximate solution does not work well when any of the efficacies are much greater than unity—which can only happen when ϕ is negative. Other reasonable approximations are possible, but were not investigated as this appears sufficiently accurate for most purposes.

Fig. 6 shows the approximate solution of efficacy for $\Phi = 0.5$ and $Z = 0.2$. The largest difference between the approximate and exact solution occurs in the region where the intermittent and asymptotical solution intersect.

6. Practical examples

The discussion above was quite general. We will now work on two practical examples from the field of ventilation. In the terminology of ventilation the solute source strength, $S(t)$, corresponds to the emission of pollutants in a room. The dilution rate, $A(t)$, is the air change rates in the room. The weighting function can be used to represent the presence of occupants in the room, so that it is possible only to evaluate the dose when the room is occupied. We assume that the dose is linearly proportional to the pollutant concentration because the vast majority of ventilation standards, are limited to chronic, long term exposure and do not address short term exposures to highly toxic substances with non-linear dose response for human health. Therefore dose is used as the metric for equivalent air quality in the following examples.

6.1. Example 1 – Intermittent ventilation with variable source generation

Consider a home ventilated at constant rate of $0.5\ \mathrm{h}^{-1}$ and assume the emission of pollutants is constant during the day. The home is occupied at all times, hence $W_1 = 1$. The owner of the house now wants to start up a business and establishes an office in the home. It is estimated that the emission of pollutants is increased by a factor of 4 during the 8 h each day that the office is used. To maintain the dose at the same pre-office level, the ventilation rate can be changed proportional to the pollutant emission rate during the office hours. Hence the rate during office hours would be $2.0\ \mathrm{h}^{-1}$ and outside office hours it would be $0.5\ \mathrm{h}^{-1}$. The amount of dilution air is therefore 24 calculated by Eq. (7). The efficacy is 1 because this is perfect dilution.

Alternatively, the ventilation could be constant during the day and adjusted to give the same dose. Because the total quantity of air over the day would be the same as the above variable ventilation rate, this new, higher, constant ventilation rate can be determined by simple averaging. The average ventilation rate that would give same dose is therefore $(16\ \mathrm{h} \cdot 0.5\ \mathrm{h}^{-1} + 8\ \mathrm{h} \cdot 2.0\ \mathrm{h}^{-1})/24\ \mathrm{h} = 1.0\ \mathrm{h}^{-1}$. The dilution is still 24, and the efficacy is 1. The difference between them is that in the *perfect dilution* case the concentration is the same for the whole period, while in the constant dilution case, the concentration varies over the period. Since we have assumed acute exposures are not an issue, these two cases are equivalent.

As an alternative to these perfect dilution cases, we could increase the ventilation rate outside office hours by 50% (from $0.5\ \mathrm{h}^{-1}$ to $0.75\ \mathrm{h}^{-1}$). Using Eq. (13) to solve for equivalent dose, we find that the rate during office hours would have be $1.25\ \mathrm{h}^{-1}$ to obtain same dose in the two systems on a daily basis. The amount of dilution air is then 22 (Eq. (7)) and the efficacy of this system compared to that of perfect dilution is 1.09. By increasing the ventilation rate outside office hours by 50% we use approximately 8% less air in total each day.

6.2. Example 2 – Demand controlled ventilation

As a first step towards evaluating a systems' energy performance, a designer wants to know the total volume of air exchanged on a daily basis in a demand controlled ventilation system (DCV) compared to a continuously operated system. The systems are to be operated in a home that is occupied for 16 h a day. Because the occupants are not present all times the systems only need to provide equivalent air quality during occupied hours. The occupied period is given the index 1 and the weighting parameters will therefore be $W_1 = 1.5$ and $W_2 = 0$. Pollutants are emitted by the building itself together with pollutants from the occupants and their activities. The emission of pollutants is assumed to be four times higher at occupied hours compared to unoccupied hours.

The continuous ventilation system is operated at an air change rate of $0.5\mathrm{h}^{-1}$ corresponding to the ventilation required in residential buildings in Denmark (see [7]). The test dilution in this system is therefore 12. Because all parameters needed to calculate that reference dilution, N_*, that gives us equivalent dose in our test and reference case of perfect dilution are given, we do not need to know the actual dilution rates in the reference system. Using Eq. (13) shows that the reference dilution is 9.9 for the continuous system.

The DCV system is operated at half rate of the continuous rate during unoccupied hours (A_2 is $0.25\ \mathrm{h}^{-1}$). We solve Eq. (13) to find the ventilation rate that during occupied hours gives us a reference dilution of 9.9. We find that this ventilation rate must be 5% higher than the rate in our continuously operated system and the test dilution, N, in the DCV case is 10.4 (Eq. (7)). Because we evaluated our two test systems relative to the same reference system, we can evaluate the two systems to each other and we find that the efficacy of the continuously operated system compared to the DCV system is 1.16. Hence, the DCV use approximately 14% less air in total every day compared to the continuously operated system to provide same indoor air quality. Further evaluations of efficacy of demand controlled ventilation with variable emission ratios are given in [15].

7. Summary and conclusions

In this paper we have derived an expression for dilution of an unsteadily-generated solute in an unsteady solvent stream, under cyclic boundary conditions. We determined an analytical relationship showing equivalence of such a process across a step-wise constant function with one step to a steady-state one. This expression was used to evaluate the efficacy i.e. how much air we needed to use in one case compared to another. Investigating the phase space of efficacy we found that a simple equation for intermittent dilution provides sufficiently accurate results at low dilution. Furthermore we found that at high dilution the efficacy approaches an asymptote. We have demonstrated how the expression can be applied to the problem of determining equivalency for different approaches to ventilation in a building where contaminants, air flows, and weightings are variable.

Acknowledgements

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Appendix A. Supplementary data

Supplementary data associated with this article can be found, in the online version, at doi:10.1016/j.ijheatmasstransfer.2010.12.018.

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Paper I - Supplementary data

"Supplementary data - Derivation of Equivalent Continuous Dilution for Cyclic Unsteady Driving Forces Supplementary data"

M.H. Sherman, D.K. Mortensen & I.S. Walker

Published online: *International Journal of Heat and Mass transf*

Supplementary data

1 Derivation of the time-varying concentration C(t)

To derive an expression for the time-varying concentration, *C(t),* we must solve the continuity equation for this. The standard integral form of an inhomogeneous, first-order, linear differential with arbitrary coefficients can be used to do this:

$$C(t) = C(t_o)\xi(t,t_o) + \int_{t_0}^{t} S(t')\xi(t,t')dt' \tag{1}$$

Where $C(t_o)$ is the known constant of integration representing the concentration at some reference time, t_o. For simplicity we have defined the following function:

$$\xi(t,t') \equiv e^{-\int_{t'}^{t} A(u)du} \tag{2}$$

Because our system is in cyclic equilibrium over the period T, some useful identities of the function are: $\xi(t,u)\xi(u,t') = \xi(t,t')$ and $\xi(t+T,t'+T) = \xi(t,t')$.

To calculate the time-varying concentration, *C(t)*, we need find the constant of integration $C(t_0)$. To do this we make us of the fact that the concentration at time zero, *C(0)*, must be the same as that at the end of the period, *C(T)*, due to our cyclic process.

$$C(0) = C(T) \tag{3}$$

$$C(t_0)\xi(0,t_0) + \int_{t_0}^{0} S(t')\xi(0,t')dt' = C(t_0)\xi(T,t_0) + \int_{t_0}^{T} S(t')\xi(T,t')dt' \tag{4}$$

Rearranging eq. 4 we find that the constant of integration is given by:

$$C(t_0) = \frac{\int_{t_0}^{T} S(t')\xi(T,t')dt' + \int_{0}^{t_0} S(t')\xi(0,t')dt'}{\xi(0,t_0) - \xi(T,t_0)} \tag{5}$$

In a first step towards simplifying the expression further we substitute, *t'*, by *u +T* and change our limits accordingly. In the denominator we make use of the fact that: $\xi(T,t_0) = \xi(T,0)\xi(0,t_0)$.

$$C(t_0) = \frac{\int_{t_0-T}^{0} S(u+T)\xi(T,u+T)du + \int_{0}^{t_0} S(t')\xi(0,t')dt'}{\xi(0,t_0)(1-\xi(T,0))} \tag{6}$$

We then make use of the fact that: $S(u+T) = S(u)$ and $\xi(T,u+T) = \xi(0,u)$ and in the end we substitute *u* by *t'*:

$$C(t_0) = \frac{\int_{t_0-T}^{0} S(t')\xi(0,t')dt' + \int_{0}^{t_0} S(t')\xi(0,t')dt'}{\xi(0,t_0)(1-\xi(T,0))} \tag{7}$$

We then expand the function $\xi(0,t')$ to $\xi(0,t_0)\xi(t_0,t')$ and make us of the fact that $\xi(0,t_0)$ is a constant we can take outside the integral.

$$C(t_0) = \frac{\xi(0,t_0)\left(\int_{t_0-T}^{0} S(t')\xi(t_0,t')dt' + \int_{0}^{t_0} S(t')\xi(t_0,t')dt'\right)}{\xi(0,t_0)(1-\xi(T,0))} \tag{8}$$

The constant of integration is then expressed by:

$$C(t_0) = \frac{\int_{t_0-T}^{0} S(t')\xi(t_0,t')dt' + \int_{0}^{t_0} S(t')\xi(t_0,t')dt'}{(1-\xi(T,0))} \tag{9}$$

The constant of integration (eq. 9) can now be put back into the solution for the concentration (eq. 1). However, because t_o is arbitrary and we have a cyclic process the constant of integration itself is a solution to the differential equation and we can remove the "0" subscript. Applying this to equation 9 and merging the integrals find that the time-varying concentration for a cyclic process can be expressed by:

$$C(t) = \frac{\int_{t-T}^{t} S(t')\xi(t,t')dt'}{(1-\xi(T,0))} \tag{10}$$

2 Derivation of an analytical expression for equivalent dose in a step wise constant process with one step

The derived expression of the dose (eq. 12 in the paper) for an unsteady but cyclic test system is compare to the dose in our reference case of perfect dilution (eq. 5 in the paper) to set up an expression for equivalent dilution:

$$C_* \, T = \frac{\oint \int_{t-T}^{t} W(t)\xi(t,t')S(t')dt'dt}{(1-\xi(T,0))} \tag{11}$$

Because we defined our time varying reference dilution rate $A_*(t)$ as that which holds the concentration constant at some steady state value, C_*, see equation 4, we do not need to individually know the solute source strength, *S(t)*, and the steady state concentration, C_{*seper}, but only the presumed dilution for perfect dilution, $A_*(t)$. Rearranging equation 11 gives us the following expression for equivalent dilution:

$$T(1-\xi(T,0)) = \oint \int_{t-T}^{t} W(t)\xi(t,t')A_*(t')dt'dt \tag{12}$$

This equation can be expanded into a sum of integrals because of the step-wise constant profiles for the solute source strength, dilution rates and weightings changing at time t_1. An analytical solution to equation 12 is found by expanding the double integral on the right side into 6 integrals each with constant weighting, solute source strength and dilution rate:

$$\begin{aligned} &T(1-\xi(T,t_1)\xi(t_1,0)) = \\ &\int_0^{t_1}\int_0^{t} W(t)\xi(t,t')A_*(t')dt'dt + \int_0^{t_1}\int_{t_1-T}^{0} W(t)\xi(t,t')A_*(t')dt'dt + \int_0^{t_1}\int_{t-T}^{t_1-T} W(t)\xi(t,t')A_*(t')dt'dt + \\ &\int_{t_1}^{T}\int_{t_1}^{t} W(t)\xi(t,t')A_*(t')dt'dt + \int_{t_1}^{T}\int_0^{t_1} W(t)\xi(t,t')A_*(t')dt'dt + \int_{t_1}^{T}\int_{t-T}^{0} W(t)\xi(t,t')A_*(t')dt'dt \end{aligned} \tag{13}$$

The analytical solution to the left side of is given by equation 14 and the 6 integrals on the right side are given in equation 15 to 20:

$$T(1-\xi(T,t_1)\xi(t_1,0)) = T\left(1-e^{-A_2(T-t_1)}e^{-A_1t_1}\right) \tag{14}$$

$$\int_0^{t_1}\int_0^{t} W(t)\xi(t,t')A_*(t')dt'dt = W_1A_{*1}\int_0^{t_1}\int_0^{t} e^{-A_1t}e^{A_1t'}dt'dt = \frac{W_1A_{*1}}{A_1^2}\left(A_1t_1 - 1 + e^{-A_1t_1}\right) \tag{15}$$

$$\int_0^{t_1}\int_{t_1-T}^{0} W(t)\xi(t,t')A_*(t')dt'dt = W_1A_{*2}\int_0^{t_1}\int_{t_1-T}^{0} e^{-A_1t}e^{A_2t'}dt'dt = \frac{W_1A_{*2}}{A_1A_2}\left(1-e^{-A_1t_1}\right)\left(1-e^{-A_2(T-t_1)}\right) \tag{16}$$

$$\begin{aligned} &\int_0^{t_1}\int_{t-T}^{t_1-T} W(t)\xi(t,t')A_*(t')dt'dt = W_1A_{*1}\int_0^{t_1}\int_{t-T}^{t_1-T} e^{-A_1t}e^{-A_2(T-t_1)}e^{-A_1(t_1-t'-T)}dt'dt \\ &= \frac{W_1A_{*1}}{A_1^2}e^{-A_2(T-t_1)}\left(1-e^{-A_1t_1} - A_1t_1e^{-A_1t_1}\right) \end{aligned} \tag{17}$$

$$\int_{t_1}^{T}\int_{t_1}^{t} W(t)\xi(t,t')A_*(t')dt'dt = W_2A_{*2}\int_{t_1}^{T}\int_{t_1}^{t} e^{-A_2t}e^{A_2t'}dt'dt = \frac{W_2A_{*2}}{A_2A_2}\left(A_2\left(T-t_1\right) + e^{-A_2(T-t_1)} - 1\right) \tag{18}$$

$$\int_{t_1}^{T}\int_0^{t_1} W(t)\xi(t,t')A_*(t')dt'dt = W_2A_{*1}\int_{t_1}^{T}\int_0^{t_1} e^{-A_2(t-t_1)}e^{-A_1(t_1-t')}dt'dt = \frac{W_2A_{*1}}{A_1A_2}\left(1-e^{-A_1t_1}\right)\left(1-e^{-A_2(T-t_1)}\right) \tag{19}$$

$$\int_{t_1}^{T}\int_{t-T}^{0} W(t)\xi(t,t')A_*(t')dt'dt = W_2A_{*2}\int_{t_1}^{T}\int_{t-T}^{0} e^{-A_2(t-t_1)}e^{-A_1t_1}e^{+A_2t'}dt'dt$$
$$= \frac{W_2A_{*2}}{A_2A_2}e^{-A_1t_1}\left(1-e^{-A_2(T-t_1)}-A_2\left(T-t_1\right)e^{-A_2(T-t_1)}\right) \tag{20}$$

The analytical equation for equivalent dose is the given by:

$$T\left(1-e^{-A_2(T-t_1)}e^{-A_1t_1}\right)=$$
$$\frac{W_1A_{*1}}{A_1^2}\left(A_1t_1-1+e^{-A_1t_1}\right)+\frac{W_1A_{*2}}{A_1A_2}\left(1-e^{-A_1t_1}\right)\left(1-e^{-A_2(T-t_1)}\right)+\frac{W_1A_{*1}}{A_1^2}e^{-A_2(T-t_1)}\left(1-e^{-A_1t_1}-A_1t_1e^{-A_1t_1}\right)$$
$$+\frac{W_2A_{*2}}{A_2A_2}\left(A_2\left(T-t_1\right)-1+e^{-A_2(T-t_1)}\right)+\frac{W_2A_{*1}}{A_1A_2}\left(1-e^{-A_1t_1}\right)\left(1-e^{-A_2(T-t_1)}\right) \tag{21}$$
$$+\frac{W_2A_{*2}}{A_2A_2}e^{-A_1t_1}\left(1-e^{-A_2(T-t_1)}-A_2\left(T-t_1\right)e^{-A_2(T-t_1)}\right)$$

This equation can be rewritten by substitution of the following variables:

Weighting: $W_1t_1+W_2(T-t_1)=T$

Reference dilution: $\mathrm{N}_*=A_{*1}t_1+A_{*2}(T-t_1)$

Actual dilution: $\mathrm{N}=A_1t_1+A_2(T-t_1)$

Fraction of time in the step: $f\equiv t_1/T$

Fractional dilution rate in the step: $r\equiv A_1T/\mathrm{N}$

Fractional source strength in the step: $r_*\equiv A_{*1}T/\mathrm{N}_*$

Substitution of the variables and grouping of term gives the following:

$$0=\frac{2e^{-\frac{N}{2}}\begin{pmatrix}\frac{1}{2}\left(-e^{-\frac{N}{2}}+e^{\frac{N}{2}}\right)Nr\left(-1+fr\right)\left(r\left(N-Nfr+N_*(-1+fr_*)\right)+N_*f\left(r-r_*\right)W1\right)\\-\frac{1}{2}\left(-e^{-\frac{Nfr}{2}}+e^{\frac{Nfr}{2}}\right)\left(e^{\frac{N(1-fr)}{2}}-e^{-\frac{N(1-fr)}{2}}\right)N_*\left(r-r_*\right)\left(r-W1\right)\end{pmatrix}T}{N^2r^2\left(-1+fr\right)^2} \tag{22}$$

Equation 22 can equivalently be expressed by the hyperbolic sine function: $\sinh\left(x\right)=\frac{1}{2}\left(e^x-e^{-x}\right)$ as:

$$0=\frac{2e^{-\frac{N}{2}}\begin{pmatrix}Nr\left(-1+fr\right)\left(r\left(N-Nfr+N_*(-1+fr_*)\right)+N_*f\left(r-r_*\right)W1\right)Sinh\left[\frac{N}{2}\right]\\-2N_*\left(r-r_*\right)\left(r-W1\right)Sinh\left[\frac{Nfr}{2}\right]Sinh\left[\frac{N(1-fr)}{2}\right]\end{pmatrix}T}{N^2r^2\left(-1+fr\right)^2} \tag{23}$$

Solving for N_* gives us the following:

$$N_* = \frac{N^2 r^2 \left(-1+fr\right)^2 Sinh\left[\frac{N}{2}\right]}{\left(\begin{array}{l} Nr\left(-1+fr\right)\left(r\left(-1+fr\right)-f\left(r-r_*\right)\left(r-W1\right)\right)Sinh\left[\frac{N}{2}\right] \\ -2\left(r-r_*\right)\left(r-W1\right)Sinh\left[\frac{Nfr}{2}\right]Sinh\left[\frac{N(1-fr)}{2}\right] \end{array}\right)} \tag{24}$$

Because $\frac{\sinh\left(A+B\right)}{\sinh\left(A\right)\sinh\left(B\right)} = \coth\left(A\right)+\coth\left(B\right)$ equation 24 can be expressed as:

$$N_* = \frac{N^2 r^2 \left(-1+fr\right)^2}{Nr\left(-1+fr\right)\left(\left(r\left(-1+fr\right)-f\left(r-r_*\right)\left(r-W1\right)\right)\right)-\frac{2\left(r-r_*\right)\left(r-W1\right)}{\left[Coth\left[\frac{Nfr}{2}\right]+Coth\left[\frac{N(1-fr)}{2}\right]\right]}} \tag{25}$$

Introducing the two variables: $\theta = f^2\left(r-r_*\right)\left(r-W1\right)$ and $Z = fr$ and rearranging equation 25 we can simplify the equation to:

$$N_* = \frac{N}{1+\frac{\theta}{(1-Z)Z}-\frac{2}{N}\frac{\theta}{(1-Z)^2 Z^2}\Big/\left[Coth\left[\frac{NZ}{2}\right]+Coth\left[\frac{N(1-Z)}{2}\right]\right]} \tag{26}$$

Paper II

'Optimization of Occupancy Based Demand Controlled Ventilation in Residences'

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Optimization of Occupancy Based Demand Controlled Ventilation in Residences

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Abstract
Although it has been used for many years in commercial buildings, the application of demand controlled ventilation in residences is limited. In this study we used occupant exposure to pollutants integrated over time (referred to as “dose”) as the metric to evaluate the effectiveness and air quality implications of demand controlled ventilation in residences. We looked at air quality for two situations. The first is that typically used in ventilation standards: the exposure over a long term. The second is to look at peak exposures that are associated with time variations in ventilation rates and pollutant generation. The pollutant generation had two components: a background rate associated with the building materials and furnishings and a second component related to occupants. The demand controlled ventilation system operated at a low air flow rate when the residence was unoccupied and at a high air flow rate when occupied. We used analytical solutions to the continuity equation to determine the ventilation effectiveness and the long-term chronic dose and peak acute exposure for a representative range of occupancy periods, pollutant generation rates and air flow rates. The results of the study showed that we can optimize the demand controlled air flow rates to reduce the quantity of air used for ventilation without introducing problematic acute conditions.

Keywords
Demand controlled ventilation, air flow rates, equivalent dose, acute to chronic exposure, effectiveness

Introduction
Ventilation is used to provide an acceptable air quality by controlling the concentration of pollutants in a space. The quantity of whole-house ventilation required to provide acceptable indoor air quality depends on the emission rates of pollutants in a space. In most buildings pollutant emission rates depend on occupancy, and are higher when occupants are present due to biological processes and occupant activities. These emissions are in addition to the emissions from materials within the building that occur independent of occupancy. Some pollutants with short emission profiles such as moisture emitted during cooking or showering are often dealt with by source control methods, although they may be considered to be a part of the background emission over the long term. Other short term emission and exposure related issues include chemical reactions and household cleaning products (Singer et al. 2006), particulate generation by cooking, particulate resuspension from vacuuming (Corsi et al. 2008), and differences in concentrations between breathing zone air and spatial averages (Novoselac et al. 2003). Codes and standards for indoor air quality in residences treat short-term high polluting localized events separately from whole house ventilation. Typically this is achieved from a standards and house design perspective

by exhausting air from kitchens and bathrooms when these rooms are in use. Other events in other rooms of a house are not explicitly addressed as there is no practical way to do so. Instead, they are dealt with indirectly through the whole house ventilation system that implicitly assumes that pollutants are well mixed in the space. This is a reasonable assumption for the long-term chronic exposures that whole house ventilation typically is recognized to address. From a practical point of view, it is also the only reasonable approach for codes, standards and system design in which spatial and temporal distribution of pollutants and the magnitude of mixing within and between zones is effectively unknowable. Disregarding these localized effects does not change the results or conclusions of this study because we are comparing the performance of whole house ventilation systems and these complications would be the same for all whole house systems. For simplicity and to ensure relevance to potential users of the equivalent dose approach, this study follows existing codes and standards for residential ventilation requirements that focus on pollutant removal by ventilation and not by other mechanisms such as filtration or sorption on surfaces. Furthermore it does not include dilution due to natural infiltration which is highly variable from building to building and with external weather conditions. Instead we focus on intentional ventilation for pollutant control.

The intent of this study is to provide results that can provide more flexible approaches to ventilation design for residences that allow Demand Controlled Ventilation (DCV) approaches to comply with codes and standards that are currently based on continuous ventilation rates. This study will also show that reductions in the quantity of air used for ventilation (and the energy used to condition this air) can be achieved without impacting health – either in terms of long-term exposure (that is addressed by current ventilation standards) or short-term acute impacts.

The ventilation required in buildings today by standards and building codes is often given by a constant air flow rate (Constant Air Volume or CAV). It is typically recognized that the rates are set to keep long-term exposures at an acceptable level. A constant ventilation rate is an appropriate solution when pollutants are emitted at a fixed rate. However, any variation in emission of pollutants means that the constant ventilation rate may lead to periods with poor short-term indoor air quality when the ventilation rate is too low and/or unnecessary energy consumption when the ventilation rate is too high. In this study appropriate ventilation rates based on demand are not set from a health perspective because thorough knowledge of all pollutants health effects on people are needed to do this. Instead we make use of the fact that the requirements for long-term acceptable air quality indirectly are set by the codes and standards. We examine the effects of varying ventilation rates as occupancy changes and look for optimum air flows that minimize the quantity of air used for ventilation that gives long-term chronic exposures equivalent to that provided by existing codes and standards. Although we do not directly calculate the energy impacts it can be assumed that reducing the quantity of air implies a reduction in energy use.

We use the concept of dose, which is the integrated exposure to a pollutant over time, as the metric for equivalent long-term chronic exposures. We assume exposure and thus dose is linearly proportional to the pollutant concentration. Dose is used because the vast majority of indoor air quality issues examined for ventilation standards are limited to chronic, long-term exposure and do not address short-term acute exposures or highly toxic substances with non-linear dose response for human health. However acute exposure can become a concern for some pollutants so we also examine the ratio of acute to chronic exposures and compare these with literature. Other criteria to assess the performance of residential ventilation systems concerning hygiene and indoor air quality are given in the standard EN15665 (CEN 2009). To provide acceptable ventilation with variable ventilation rates we require that the dose be the same or lower than that provided by a constant ventilation rate.

Occupants are not exposed to pollutants when they are absent and a key concept in this study is therefore to limit dose calculations to times when occupants are present. Our task is to identify the air flow rates that provide the same dose. Because concentration and ventilation are dynamically and inversely related through the continuity equation the dose cannot be calculated in a straightforward manner. Instead, we develop analytical solutions that specify how much air is needed in one ventilation system compared to another to obtain the same dose. We define the ratio of air requirements between systems as the ventilation effectiveness.

Background

The principle of air quality equivalency in terms of dose was studied for intermittent ventilation systems by Sherman (2006). The results of the study have been included in ASHRAE Standard 62.2 (ASHRAE 2010) by allowing intermittent ventilation provided that the ventilation rate is raised when the ventilation system is operating. Sherman's study was limited to on/off operation of the ventilation system, constant emission of pollutants, and dose was evaluated on a 24 hour basis.

Sherman expanded the study of equivalent air quality in terms of dose so that the three parameters: ventilation rates, emission rates and the evaluation period of dose could vary (Sherman et al. 2011). Because roughly the same things occur in a building on a daily basis the pollutant emission and ventilation patterns are repeated resulting in a cyclic pollutant concentration and a general expression for dilution of an unsteadily generated pollutant by a variable ventilation rate, under cyclic temporal boundary conditions was derived.

The general equation for the time-varying concentration under cyclic boundary conditions is:

$$C(t) = \frac{\int_{t-T}^{t} S(t')\xi(t,t')dt'}{(1-\xi(T,0))} \tag{1}$$

Where

$$\xi(t,t') \equiv e^{-\int_{t'}^{t} A(u)\,du} \tag{2}$$

C is pollutant concentration, *A* is ventilation rate, *S* is pollutant source strength, *T* is the duration of the cyclic period and *t* is time. The time-varying concentration was integrated over the cyclic period *T* to calculate the dose *d* (eq. 3). To omit or emphasize parts of the cyclic period differently than others a weighting function *W* was added. The weighting function can account for occupancy, i.e., when occupants are absent *W*=0 and the pollutant concentration during that period do not contribute to the dose.

$$d = \frac{\oint W(t) \int_{t-T}^{t} S(t')\xi(t,t')dt'dt}{(1-\xi(T,0))} \tag{3}$$

Method

These equations were used in this study to calculate the ventilation effectiveness of a DCV system together with the system's effect on indoor air quality. The effectiveness only considers time variation of the air flow rate and not local inefficiencies associated with imperfect mixing within and between zones or the spatial distribution of pollutants in the home. Pollutants were assumed to be removed by ventilation and not by other mechanisms such as filtration or sorption on surfaces. The performance was evaluated using a CAV system as a reference case and this system set the target for equivalent dose. The performance of the DCV system was evaluated assuming a repeated 24 hour cycle during which there is one step change in pollutant emission rates from high to low corresponding to a change from the residence being occupied to unoccupied. There is a corresponding step up at the end of the unoccupied period. There is only one occupied period in each 24 hour cycle. During both occupied and unoccupied times the pollutant emission and ventilation rates are constant. Because the ventilation and emission profiles were step-wise constant we could set up an analytical expression for equivalent dose for the CAV and DCV systems using Equation 3. The equivalency equation was solved to find the air flow rates in the DCV system that provided equivalent dose to the CAV system.

The generation of pollutants comprised of a constant part ($S_{constant}$) associated with the building and an intermittent part ($S_{intermittent}$) associated with the occupants. The pollutants were assumed to be additive resulting in a step-wise constant emission profile. The pollutant profile was described by the emission ratio (*ER*) relating the emission during occupied hours to unoccupied hours.

$$\mathrm{ER} = \frac{\mathrm{S_{constant}} + \mathrm{S_{intermittent}}}{\mathrm{S_{constant}}} \tag{4}$$

The DCV system was controlled by occupancy with a high air flow rate during occupied hours and a low air flow rate during unoccupied hours. There exist many combinations of high and low flow rates that provide a dose equivalent to that in the CAV system. However, the range of possible DCV systems is restricted by the low rate ($A_{DCV,low}$) that never can be less than zero. We further limited our investigations to DCV systems where the upper bound for the low rate is the ventilation rate of the CAV system. The low rate was therefore used to categorize the range of DCV systems by introducing the Low-Ventilation Factor (*LVF*) that expressed the low ventilation rate as a percentage of the CAV rate, A_{CAV}. At a low-ventilation factor of 1 the low and high air flow rates are identical.

$$\mathrm{LVF} = \frac{\mathrm{A_{DCV,low}}}{\mathrm{A_{CAV}}} \tag{5}$$

The low and high ($A_{DCV,high}$) air flow rates that provided equivalent dose were used to express the effectiveness (ε) of the system. The effectiveness is a measure of how good the DCV system is at providing an air quality relative to the CAV case. The effectiveness is defined by the volume of air one would need in the reference system to that needed in the DCV system throughout the cyclic period. For the occupancy controlled DCV system the effectiveness is calculated by:

$$\varepsilon = \frac{\mathrm{A_{CAV}}}{\mathrm{A_{DCV,low}} \cdot (1 - \mathrm{f_{occ}}) + \mathrm{A_{DCV,high}} \cdot \mathrm{f_{occ}}} \tag{6}$$

Where f_{occ} is the fractional occupied time during the cyclic period *T*. Systems can have equivalent dose but different cyclic concentration profiles resulting in different peak concentrations. To evaluate the overall air quality performance of the systems we calculated acute to chronic exposures represented by peak to average dose exposures using Equations 1 and 3. Furthermore, an analysis determining how uncertainties in the high and low ventilation rates influence the effectiveness and dose during occupied hours was made.

Example Calculations

To determine optimum air flow rates for occupancy controlled DCV systems the total air flow reductions over 24 hours were calculated for three scenarios using representative values for CAV air flow rate, occupied hours and emission ratios.

In the first scenario we evaluated the effect of increasing the ventilation rate when people are present. The generation of pollutants was constant (ER=1) and we used a reference CAV rate of $0.5h^{-1}$. The number of occupied hours was based on studies of occupancy in buildings (Brasche et al. 2005, Leech et al. 2002) that showed that people in general spend 16 hours a day in their home. To cover upper and lower limits of people's presence in their home occupancies of 8 and 20 hours were also analyzed.

The second scenario evaluated the effect of increasing the ventilation rate when people are present and more pollutants are emitted during these hours. We used a reference CAV rate of $0.5h^{-1}$ and assumed people were present in the home 16 hours a day. Emission ratios were deduced from ASHRAE standard 62.2 (ASHRAE 2010) and EN15251 (CEN 2007) that both use floor area and number of occupants to specify continuous ventilation requirements. The floor area is related to the emission of pollutants from the building and the number of occupants is related to the additional emission of pollutants due to occupants. In this study we assumed that pollutant emission rates are proportional to the air flow rates in the standards. The emission ratios for a home of $120m^2$ and varying number of occupants are given in Figure 1(a). A common occupancy for the home is estimated to be 2-3 people, which means that ER equals approximately 1.5. In Figure 1(b) minimum, maximum and mean emission ratios for homes of 60, 120 and $180m^2$ and expected occupancies have been calculated from the two standards. The average value for all three homes is approximately 1.5 and we evaluate the effectiveness using this value.

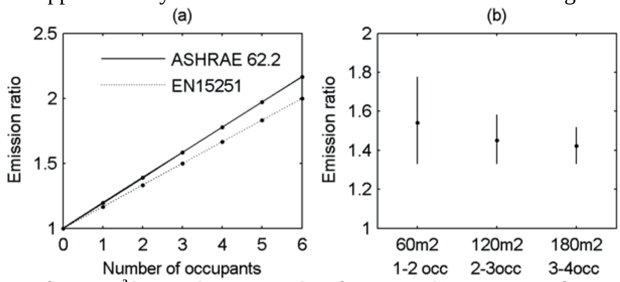

Figure 1: (a) Emission ratios for a $120m^2$ home with various number of occupants. (b) Emission ratio for typical matching home sizes and occupants.

Furthermore we analyzed cases with upper and lower limits of ER equal to 1 and 4. The emission ratio of 1 corresponds to occupants emitting no pollutants. The emission ratio of 4 corresponds to people being the main pollutant source. This high ER case is of increasing interest as occupant generated pollutants becomes more important due to the development, regulation and labeling (e.g., California Environmental Protection Agency composite wood product Airborne Toxic Control Measure (CEPA 2011) and Danish Indoor Climate Labeling (DICL 2011)) of low emitting buildings materials and furniture.

The last scenario evaluated different reference CAV rates. We did this for a case with 16 occupied hours and an emission ratio of 1.5. The CAV air flow rates were selected based on residential ventilation requirements. The ventilation required in residential buildings in Denmark (BR10 2010) corresponds to $0.5h^{-1}$. The ventilation required by ASHRAE 62.2 is approximately $0.35h^{-1}$ including a credit for infiltration and we use this as a lower boundary for the CAV rate. Furthermore we analyzed at an upper limit for the CAV rate of $1.0h^{-1}$.

Results

Cyclic concentration profiles

To enhance the explanation of the results of dose based design of DCV systems, the 24 hour cyclic concentrations for low-ventilation factors of 1, 0.75, 0.5, 0.25 and 0 and emission ratios of 1 and 4 are shown in Figures 2 and 3, respectively. Both figures have a reference CAV rate of $0.5h^{-1}$ and 16 occupied hours (hour 0 to 16). The cyclic concentration is normalized to the maximum concentration for the CAV system. Integration of the cyclic concentration from hour 0 to 16 gives us the occupant dose. The occupant dose is equivalent for the five low-ventilation factors when the emission ratio does not change. However, the dose changes when the emission ratio, reference CAV rate or the number of occupied hours change. At an emission ratio of 1 the CAV rate holds the concentration constant at a steady state value shown in Figure 2. Lowering the ventilation rate during unoccupied hours results in increased concentration at the beginning of the occupied period and the peak concentration in the DCV systems is therefore always higher than the peak concentration in the CAV system.

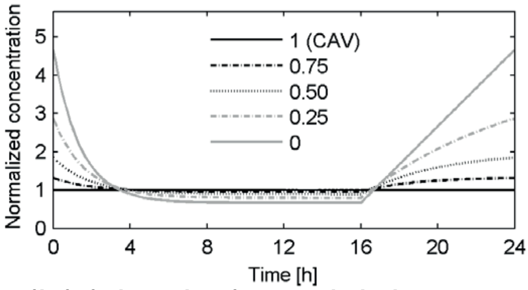

Figure 2: Cyclic concentration profiles for five low-ventilation factors normalized to the maximum concentration in the reference CAV system. All systems have an emission ratio of 1, 16 occupied hours and the reference CAV rate is $0.5h^{-1}$.

Changing the emission ratio from 1 to 4 strengthens the incentive to ventilate less during unoccupied periods. When the high to low air flow ratio equals the emission ratio the concentration is held at a constant steady state value and for ER=4 this occurs when LFV is between 0.25 and 0.5, as shown in Figure 3. At high to low air flow ratios above ER the peak concentration occurs at the beginning of the occupied period but shifts to the end of the occupied period when the high to low air flow ratio is below the ER.

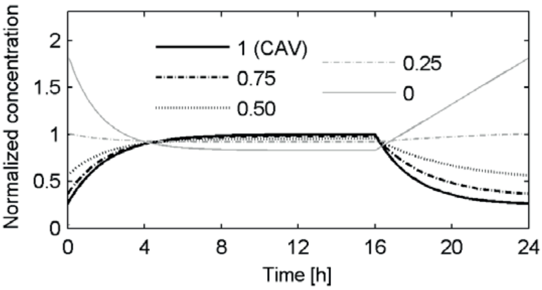

Figure 3: Cyclic concentration profiles for five low-ventilation factors normalized to the maximum concentration in the reference CAV system. All systems have an emission ratio of 4, 16 occupied hours and the reference CAV rate is $0.5h^{-1}$.

Scenario 1

Effectiveness curves for 8, 16 and 20 occupied hours are given in Figure 4. Each occupancy time has one combination of low and high flow rates where the effectiveness peaks. This peak corresponds to the minimum amount of air required to provide equivalent dose. The effectiveness is 1 at the LVF boundaries of 0 and 1. The upper LVF boundary is identical to the CAV that is our reference case. At the lower boundary where there is no ventilation during unoccupied hours we observe that ventilation is linearly related to concentration for ER=1. The maximum effectiveness increases with fewer occupied hours from 1.03 to 1.36 within a narrow range of LVF from 0.33 to 0.44. For the case of 16 occupied hours and a CAV rate of $0.5h^{-1}$ the amount of air can be reduced by 9% (maximum effectiveness=1.10) when the LVF is 0.4.

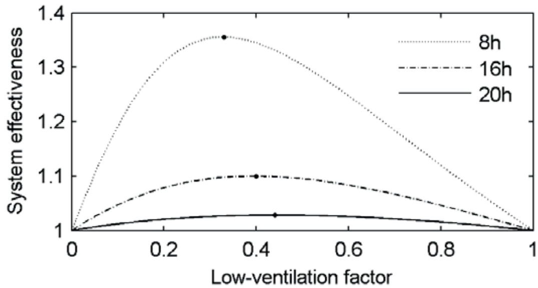

Figure 4: Changes in effectiveness for three periods of occupancy as a function of the low-ventilation factor. Reference CAV rate is $0.5h^{-1}$ and emission ratio is 1. Peak effectiveness' are marked by a dot.

Table 1 show the high and low air flow rates necessary to achieve maximum effectiveness expressed as a percentage of the CAV rate (in the same way as the low-ventilation factor expresses the low air flow rate as a percentage of the CAV rate). Furthermore the high to low are flow ratios are calculated. Shorter occupied times require higher air flow rates during occupied hours to provide equivalent dose but also lower air flow rates during unoccupied hours. This result in increased high-to-low air flow ratios with fewer occupied hours.

Table 1: High and low air flow factors at peak effectiveness.

	8h	16h	20h
LVF ($A_{DCV,low}$/ A_{CAV})	0.33	0.40	0.44
$A_{DCV,high}$/ A_{CAV}	1.55	1.16	1.08
High-to-low air flow ratio	4.7	2.9	2.5

Figure 5 shows the variability in acute to chronic exposure. Because the CAV rate holds the concentration constant at a steady state level the acute to chronic exposure is 1 at LVF=1 and above 1 for all other LFV and the peak concentration always occurs at the beginning of the occupied period (see cyclic concentration profiles for ER=1 in Figure 2). For the case of 16 occupied hours the acute to chronic exposure at maximum effectiveness (LVF=0.40) is approximately 2.2 times that in the CAV system. The acute to chronic exposure is 1.7 to 2.9 at maximum effectiveness for occupancies of 20 and 8 hours respectively.

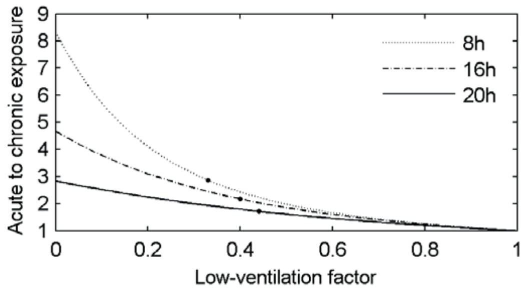

Figure 5: Acute to chronic exposure for 8, 16 and 20 occupied hours. Peak effectiveness' are marked by a dot on the curves.

Scenario 2

The second scenario evaluated the effect of increasing the ventilation and pollutant emission rate when occupants are present for the case where occupants are present 16 hours a day and the reference CAV rate is $0.5h^{-1}$. Figure 6 shows effectiveness curves for emission ratios of 1, 1.5 and 4 where the peak effectiveness is 1.10 to 1.22. These maximum values occur when the LVF is in the range of 0.13 to 0.4. The greatest reduction in total volume of air is 18% (maximum effectiveness=1.22) when the ER is 4.

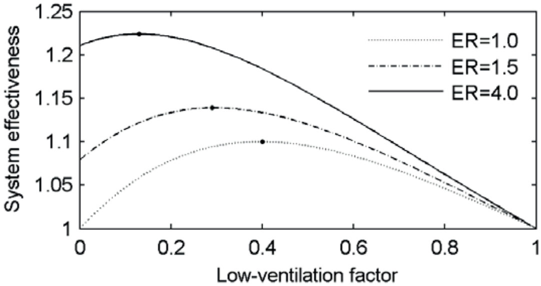

Figure 6: Changes in effectiveness for three emission ratios as a function of the low-ventilation factor. Peak effectiveness' are marked by a dot.

The high and low air flow factors necessary to achieve maximum effectiveness expressed as a percentage of the CAV rate are given in table 2 together with the high-to-low are flow ratios. The high air flow rate at peak effectiveness is almost independent of the emission ratio. However the low-ventilation factor is reduced with higher emission rates resulting in increased high-to-low air flow ratios at higher emission ratios.

Table 2: High and low air flow factors at peak effectiveness.

	ER=1	ER=1.5	ER=4.0
LVF ($A_{DCV,low}$/ A_{CAV})	0.4	0.29	0.13
$A_{DCV,high}$/ A_{CAV}	1.16	1.17	1.16
High-to-low air flow ratio	2.9	4.0	8.9

Figure 7 shows how the acute to chronic exposure changes with emission ratio. At LVF=1 the acute to chronic exposure is above 1 when ER>1 (1.04 when ER=1.5 and 1.10 when ER=4) because the cyclic concentration is not steady, whereas it is 1 when ER=1. For ER=1.5 the acute to chronic ratio is 1 when LFV is 0.70 because the high to low ventilation ratio is 1.5 (high-ventilation factor=1.04) and the concentration is thereby held at constant steady state value. In this system the effectiveness is 1.08 resulting in a 7% reduction in total volume of air. When ER=4 the acute to chronic exposure is 1 when LFV=0.28 and the high-ventilation factor is approximately 1.10. The effectiveness is 1.21 resulting in a reduction in total volume of air of 17%. This means that if ER>1 we can reduce the total volume of air and at the same time improve the air quality compared to CAV operation. The acute to chronic ratio is between 1.3 and 2.2 at maximum effectiveness. The highest ratio and thereby the worst case occurs when the emission ratio is 1.

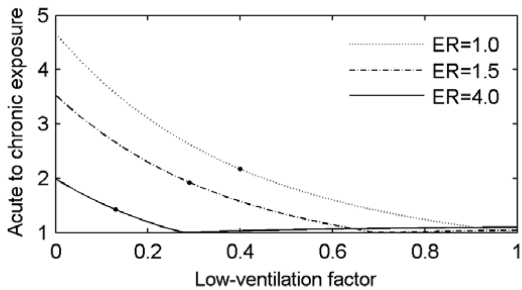

Figure 7: Acute to chronic ratio as a function of low-ventilation factor. Peak effectiveness' are marked by a dot on the curves.

Scenario 3

Finally we evaluate the reference CAV rates effect on system effectiveness. We do this for the case where people are home 16 hours a day and the emission ratio is 1.5. Figure 8 shows the effectiveness for CAV rates of 0.35, 0.5 and $1.0h^{-1}$ and it is seen that the maximum effectiveness increases with increasing CAV rate. The maximum effectiveness range from 1.10 to 1.21 and at these peak values the low-ventilation factor is 0.32 and 0.24 respectively. At a CAV rate of $0.5h^{-1}$ the maximum expected reduction in total volume of air is about 12% (maximum effectiveness=1.10 at LFV=0.29).

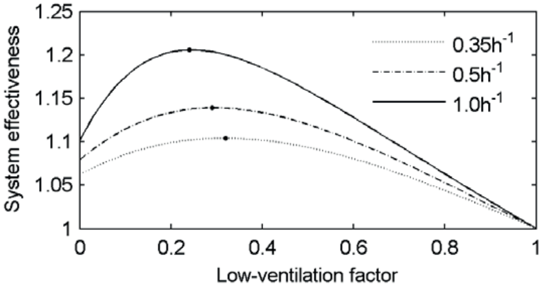

Figure 8: Changes in effectiveness for CAV rates of 0.35, 0.5 and $1.0h^{-1}$. Peak effectiveness' are marked by at dot.

The high and low air flow rates necessary to achieve maximum effectiveness expressed as a percentage of the CAV rate are given in table 3 together with high-to-low flow ratios. Higher reference CAV rates are more effective in removing pollutants; hence the air flow factor is lower at both occupied and unoccupied times than systems with low reference CAV rates. However the high-to-low ratio increases with higher reference CAV rate.

Table 3: High and low air flow factors at peak effectiveness.

	$0.35h^{-1}$	$0.5h^{-1}$	$1.0h^{-1}$
LVF ($A_{DCV,low}$/ A_{CAV})	0.32	0.29	0.24
$A_{DCV,high}$/ A_{CAV}	1.20	1.17	1.12
High-to-low air flow ratio	3.7	4.0	4.7

The acute to chronic ratios for CAV rates of 0.35, 0.5 and 1.0 are given in figure 9. The ratio equals 1 when the high to low ventilation ratio is 1.5 but because the reference CAV rate affects the pollutant accumulation rate the low-ventilation factor at the steady state concentration will not be the same. LVF is approximately 0.69 to 0.71 for the three CAV rates. At maximum effectiveness the acute to chronic ratio is 1.7 to 2.6 with highest values at high CAV rates.

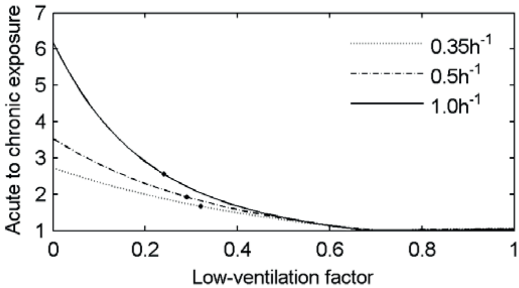

Figure 9: Acute to chronic exposure as a function of the low-ventilation factor. Peak effectiveness' are marked by a dot on the curves.

Implications of uncertainty in ventilation rates

Figure 10 shows ventilation effectiveness and dose during occupied hours relative to their values at peak effectiveness for uncertainties in high and low ventilation rates of ±5%. Changes in dose during occupied hours are calculated by eq. 3 for the cases included in the three scenarios. Changes in effectiveness are calculated by eq. 6.

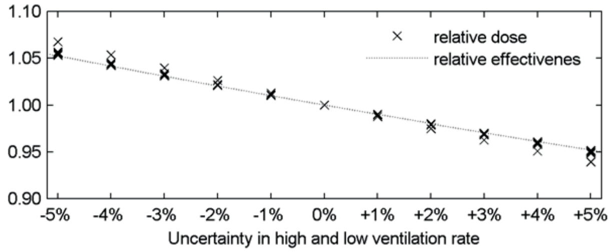

Figure 10: Effectiveness and dose during occupied hours relative to their values at peak effectiveness for ±5% changes in ventilation rates

The relative effectiveness and dose are both approximately inversely proportional to the uncertainty in ventilation rates. Uncertainties of ±5% in high and low ventilation rates result in uncertainties of ±5% in effectiveness and dose relative to their values at peak effectiveness.

Discussion

The results show that the performance of a DCV system can be optimized given occupancy time and emission ratio. Despite the variation of the parameters the three scenarios have many common characteristics. Firstly all values of peak effectiveness lie within a limited range from 1.03 to 1.36. Furthermore none of the investigated cases had an effectiveness below 1. This means that we can expect reductions in total volume of air up to 26% by redistributing the air to times of occupancy and never use more air than in our reference CAV case. We have thereby demonstrated an upper limit to the theoretically expected reductions. A reasonable estimate of the expected reduction in total volume of air is 12% representing the case of 16 occupied hours, a reference CAV rate of $0.5h^{-1}$ and an emission ratio of 1.5.

Another common characteristic is that the low-ventilation factor was 0.13 to 0.4 at peak effectiveness. This means that peak effectiveness occurred when the low air flow rate was 13% to 40% of the reference CAV rate independent of occupancy, emission ratio and reference CAV rate. At peak effectiveness the high air flow rate ranged from 108% to 154% of the reference CAV rate. By pairing the flow rates that provide equivalent dose, the high to low air flow ratio ranged from 2.5 to almost 9. This ratio is of interest when sizing ducts and selecting fans. The largest differences in high to low air flow ratio occurred in the system with 16 occupied hours, a reference CAV rate of $0.5h^{-1}$ and ER=4. This change in flow ratio was primarily due to a reduced low air flow rate. All other systems had a high to low air flow ratio of 2.5 to approximately 5 at peak effectiveness.

The relative uncertainty in effectiveness and dose during occupied hours compared to their values at peak effectiveness were approximately inversely proportional to the uncertainty in ventilation rates. An uncertainty of ±5% in high and low ventilation rates translate to a similar uncertainty in the predicted relative dose and relative effectiveness. Specific pollutants must be addressed to determine if such changes in dose are acceptable since it differs from that obtained when meeting standards and codes requirement to ventilation. Similar uncertainty analysis could be made for the emission ratio, reference CAV rate and the number of occupied hours.

A significant consequence associated with dose based design of a DCV system is that the peak concentration changes. The metric, acute to chronic exposure was used to evaluate this effect. At maximum effectiveness the highest acute to chronic ratio was below 3. To determine if peak concentrations are an issue of concern we need to look at the differences between chronic long-term and acute short-term health effects. A literature review of reported chemical pollutants in residences indentified 23 pollutants of concern as chronic hazards (Logue et al. 2010). The acute to chronic ratio for these pollutants was determined based on published health standards (Sherman et al. 2010). The health standards based short-term exposures on 1, 8 or 24 hour averaged values whereas our peak concentration was an instantaneous value. Averaging of our peak concentration over 1 or more hours will therefore lead to lower acute to chronic ratios. The pollutants with the lowest acute to chronic ratios were PM2.5, NO_2 and formaldehyde with ratios of 2.5 (24h average), 5.4 (8h average) and 4.7 (1h average) respectively. Because outdoor air can be a significant source of particulates we used formaldehyde as the limiting case. Therefore, if the ratio of the acute to chronic exposure in our DCV systems is below 4.7 then the peak concentrations are acceptable. As our results showed, the ratio is always less than 3, meaning that the peak concentrations are acceptable and not a barrier to adoption of the DCV technique in residential applications. The results also showed that if occupants contribute to the majority of emissions then acute to chronic ratios may be lower for the DCV system than for a CAV system. In the limit we only need to ventilate when the home is occupied.

Conclusions

Theoretical evaluations of effectiveness of occupancy controlled ventilation compared to CAV operation were carried out. The evaluations were based on a range of assumptions e.g. the ventilated space was perfectly mixed, different pollutants' load could be added, and the hours of occupancy were fixed and consecutive. The results provide an estimate of the expected impact of DCV in residential buildings but because of the assumptions the results are not necessarily applicable outside that range and not definitive in the real world. However it was evident that if you know when occupants are present a DCV system can reduce the air necessary to achieve acceptable indoor air quality. For a home occupied 16 hours a day reductions in total volume of exchanged air is about 12%. For a limiting case of no occupant contribution to pollutants, the reduction is about 9%. At the other extreme of occupant dominated pollutant emissions the reductions are 18% or more. The trade off is an increase peak concentration. However, the increase in acute to chronic exposure is well below the acute to chronic exposures of concern derived from health standards.

Acknowledgement

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Paper III

'System design for demand controlled ventilation in residential buildings'

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System design for demand controlled ventilation in multi-family dwellings

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1 Keywords
Residential ventilation, demand controlled ventilation, control strategy, static pressure reset, energy consumption

2 Abstract
This paper investigated solutions for system design of a centralized DCV system in multi-family dwellings. The design focused on simple and inexpensive solutions. An economical estimate showed that the initial cost of implementing DCV in a system with an efficient heat exchanger should not exceed 3400 DKK per dwelling. A design expected to fulfill this requirement was investigated in detail with regard to its electricity consumption by evaluation of different control strategies. Systems with variable air flows are typically controlled by maintaining the static pressure at a fixed level at a selected point in the main duct. However, sustaining the static pressure at a fixed level at part load leads to throttling of all control components and thereby unnecessary energy consumption. Resetting the static pressure at part load reduces throttling and energy can be saved. A static pressure reset strategy was applied to a dwelling-specific DCV system where the air flow varied between three fixed rates. The systems performance was evaluated for two diffusers. The annual electricity consumption was reduced by 20% to 30% when resetting the static pressure at part load condition compared to a control strategy with fixed static pressure.

3 Introduction
Ventilation of residential buildings is needed to provide a good indoor air quality and avoid deterioration of the building structure. This need has mainly been accommodated by intentional openings and infiltration through leaks in the building envelope often assisted by mechanical exhaust from the bathroom and kitchen. However, the requirement for improved energy performance of buildings has resulted in tighter building envelopes and air must be supplied and exhausted actively. This is i.e. substantiated by measurements of the CO_2 level in a newly built tight home that rose significantly when the mechanical ventilation system was turned off (Nielsen et al. 2009). A balanced ventilation system where the air flow is controlled according to the level of pollutants ensures a good air quality and is likely to be more energy efficient than a system with constant air flow because the air flow rate can be reduced at periods of low demand. Systems with variable air flows (to manage air quality or space conditioning loads) are mainly known from offices and schools. These systems often have a high initial cost because adjustable dampers and/or air terminal devices are needed to control the air flow but their operational cost is lower than constant air volume (CAV) systems due to lower average outdoor air flows. Ventilation requirements in residential buildings are significantly smaller than in schools and offices. The minimum required air change rate in Danish residential buildings is 0.5 h^{-1} (0.30 l/s m^2 (net area)) (BR10 2010) whereas the recommended ventilation rates in landscaped offices and class rooms are 1.2 l/s m^2 and 4.2 l/s m^2, respectively, according to EN15251 (CEN 2007) for low a polluting building and indoor climate class II. The energy saving potential in residential buildings is therefore substantially reduced and the

system design should be inexpensive to be cost-effective. A simple and cost-effective solution for system design of a centralized DCV system in multi-family dwellings is proposed in this paper. The centralized system is equipped with an efficient heat exchanger and possible savings on heating due to air flow reductions are thereby significantly reduced. Focus is therefore on the control of the system and the energy consumption associated with this. The developed solution's fan power consumption and expected energy savings are analyzed for different control strategies and two types of diffusers.

4 Variable air flow systems

Demand Controlled Ventilation (DCV) is a method to maintain a certain indoor air quality (IAQ) level by adjusting the air flow according to the demand. Reductions of 5% to 30% on average air flow/ventilation heat loss are reported for residential DCV systems (Mansson et al. 1993, Afshari et al. 2005). The fundamental function of a DCV system is to vary the air flow and it is therefore natural to look at operation and control of Variable Air Volume (VAV) systems. Local control components i.e. dampers and/or air terminals are used to vary the flow by decreasing or increasing the mechanical energy losses along the flow path. To provide stability and ensure correct distribution of the air, the static pressure is usually maintained at a fixed level at a selected point in the main duct. It is recommended to locate the sensor 75% to 100% of the distance from the first to the most remote terminal (ASHRAE 2007). The most energy efficient way to maintain the static pressure at the fixed set point is to alter the speed of the fan by a Variable Speed Drive (VSD). However, the full energy saving potential is not obtained because pressure must be throttled off at part load conditions to maintain the static pressure at the fixed level. The degree of throttling in a ventilation system should be as low as possible to reduce fan energy consumption and avoid problems with noise (Sørensen 2002). Resetting the static pressure set point at part load reduces throttling and energy can be saved. The reset value should be high enough to avoid that zones are starved of airflow and low enough to keep the damper along the critical path fully open to avoid throttling. The critical path in a system with variable air flow rates is dynamic and continually changes as loads in the building change (ASHRAE 2009) and a static pressure reset (SPR) control method needs to identify these changes. Some methods use control component position or a saturation signal (derived from e.g. the air flow through or the position of the local control component) to identify changes and generate a pressure request while others use predefined empirical reset schedules or a calibrated model of the ventilation system to make instantaneous calculations of the critical pressure (ASHRAE 2007). The saturation signal method can use dampers where the actual position is unknown but only indicates if it is full open. The 'trim and respond' strategy analyzed by Taylor (2007) uses a saturation signal to set up an algorithm that stepwise increases (respond) or decreases (trim) the static pressure until one damper is fully open. The strategy is appealing due to its intuitive tuning parameters and because it reacts to the actual condition in the system as it is a closed control loop. A linear pressure reset schedule based on supply air flow was developed by Federspiel (2003). The strategy correlate supply air flow and static pressure. Therefore, the strategy does not produce optimal performance but is appealing because it is simple to implement and stable due to the open control loop. This strategy resulted in fan power savings of 26% compared to fixed static pressure control. Other reported fan energy savings range from 30% to 50% compared to fixed pressure setpoint control (ASHRAE 2007).

5 System design specifications

We want to employ a SPR control strategy in design solutions for DCV in multi-family dwellings. Before specific solutions are considered the performance requirements of the system are specified. Our objective is to develop a simple, inexpensive and energy efficient centrally balanced mechanical ventilation system for multi-family dwellings.

The air flow to each dwelling should be individually controllable. Each dwelling has a least one sensor to indicate ventilation demand and one air flow control component. A pressure sensor is located in the main duct to control the speed of the fan. This means that flow rates are controlled on *system* level by the fan and on *local* level by dampers and/or diffusers. The control system should ensure balance in air supply and exhaust in each dwelling (Svensson 2008).

The location and properties of the diffusers impact the indoor environment. The thermal environment is mainly influenced by the inlet diffuser through the supply temperature and air velocities. Draught and dumping of cold air should be avoided. The location of the diffusers impacts the atmospheric environment. Air should be supplied to living areas and exhausted from high pollutant rooms (e.g. kitchens and bathrooms). This configuration provides an air flow pathway that is beneficial for the occupants and the durability of the dwelling. Nor should the diffusers produce disturbing noise. The air flow to a room should not vary more than ±10% on supply flow and ±15% on exhaust flow (DS 2005).

The specific fan power (SFP) must not exceed 2100 J/m^3 at maximum flow which is the present requirement in the Danish Building Regulation for balanced ventilation system with variable flow rates (BR10 2010). Extensive energy use in the duct system is avoided when the ducts are sized to a resistance below 1 Pa/m.

Air flow rates are set based on theoretical calculations of occupancy based DCV systems that comply with a continuous whole house ventilation requirement of 0.5 h^{-1} with regard to long-term chronic exposures (Mortensen et al. 2011). With a typical occupancy of 16 h per day in homes and a moderate change in pollutant emission during occupied and unoccupied hours the air flow rate can be reduced to 29% of the CAV rate during unoccupied hours and increased to 117% of the CAV rate during occupied hours without introducing problematic short-term acute conditions. This setup exchange less air on a daily basis compared to the CAV system. The air flow rates are approximately 0.15 h^{-1} and 0.6 h^{-1} (or 0.1l/s pr. m^2 and 0.40 l/s m^2) during unoccupied and occupied hours respectively. We use the air flow rate during unoccupied hours as the lower bound in our analyses. This is comparable to the recommended minimum air flow rate (0.05 l/s m^2 to 0.1l/s m^2) given by standard EN15251 (2007) if no value is set on national level. The upper limit is set to approximately double the basic rate (1.2 h^{-1} or 0.8 l/s m^2) which is comparable to the air flow exhausted from an existing Danish home when the kitchen hood is operated.

The initial cost of the system should not exceed the net present value (eq. 1) of the expected savings by implementing DCV during the lifetime of the system.

$$Net\ present\ value = \sum_{t=1}^{life\ time} \frac{Annual\ saving}{(1+interest\ rate)^t} \quad (1)$$

The net present value of the additional savings due to DCV in a dwelling of 70m^2 already equipped with a heat exchanger is calculated to estimate the cost-effective initial cost. The constant air change rate in the dwelling is 0.5 h^{-1} before DCV is implemented. The ventilation system has a SFP value of 1200 J/m^3 and the heat exchanger has an efficiency of 80% corresponding to an efficient system today. The energy consumption for heating the ventilation air is calculated based on a climate with 3000 heating degree days approximately equivalent to the Danish climate. Heating and electricity savings associated with DCV are assumed to be proportional to the reported heat/air flow reductions for residential DCV systems. We use a reduction of 30% to estimate the most favorable present value of the savings. The expected annual heating and electricity expenses and the annual savings are given in table 1 with heating and electricity costs of 0.4 DKK/kWh and 2 DKK/kWh respectively.

Table 1: Estimated annual heating and electricity expenses and savings on ventilation in a dwelling with and without DCV

	Annual heating expenses	Annual electricity expenses
CAV	424 kWh (170 DKK)	256 kWh (512 DKK)
DCV	297 kWh (119 DKK)	179 kWh (358 DKK)
Annual saving	127 kWh (51 DKK)	77 kWh (154 DKK)

With an expected lifetime of 20 year for the ventilation system and an interest rate of 2% the net present value is approximately 3400 DKK per dwelling. The initial cost for the developed solutions should not exceed this value to be cost-effective.

5.1 Generic system design

The specified performance requirements regarding air flow control allows for two DCV approaches: dwelling-specific flow control or room-specific flow control. Generic system designs are seen in figure 1.

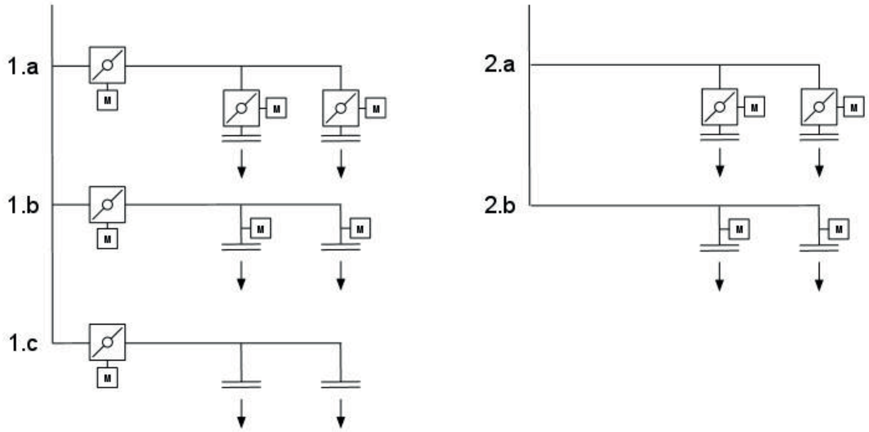

Figure 1: Generic layouts of system design for dwelling-specific and room-specific DCV

The simplest layout is system 1.c where a single damper controls the air flow to each dwelling. In this design it is not possible to control the air flow individually to each room. In order to have air flow control for each room it is necessary to apply motorized dampers or motorized inlet diffusers in each room. Systems 1.a, 1.b, 2.a and 2.b show layouts suitable for individual room control. Motorized inlet diffusers (system 1.b and 2.b) make it possible to maintain a constant throw by varying the inlet area. Systems with fixed diffusers will have a variable throw, and must be designed to avoid draught and to ensure reasonable mixing.

5.2 Cost-effective system design

The initial cost of a system increases with the number of control components. Furthermore, more control components result in more data signals to manage and maintain and thereby higher cost and complexity. Also the energy to operate more control components per dwelling can become a substantial part of the total running cost. The initial cost of a room-specific DCV system will exceed the estimated net present value of the savings (estimated to 3400 DKK) and is therefore not expected to be cost effective in multi-family dwellings at the moment. The performance of the dwelling-specific DCV layout (system 1.c) is therefore analyzed further with regard to its control strategy and diffuser types.

The occupant density in dwellings is typically lower and less variable than in non-residential buildings and occupancy detection is therefore estimated to be an acceptable and inexpensive control variable for DCV in dwellings. Therefore the air flow to each dwelling is varied between three fixed rates. A low air flow to dilute constantly emitted pollutants when people are absent denoted 'q_L'. A basic flow to dilute occupant associated pollutants denoted 'q_B' and a forced flow denoted 'q_F' to dilute or remove pollutants associated with activities such as cooking, showering etc.

The SPR strategy is applied to the dwelling-specific DCV systems where the air flow varies between three fixed rates. We first analyze the control strategies that set the boundaries of the annual electricity consumption: a control strategy with fixed static pressure sets the upper boundary and a closed loop control with ideal reset of the static pressure sets the lower boundary. Then we analyze an open control loop that uses predetermined damper positions and a static pressure reset schedule based on the air flow to the dwellings to control the flow.

6 Analyzed system

A building with 6 dwellings of 70m^2 was the basis for the analyses. Each dwelling has a bedroom, a living room, a bathroom and a kitchen. The minimum air flow rate to each dwelling is 7 l/s and the maximum flow is 56l/s. The air flow at basic level is 28l/s. The program PFS (PFS 2000) was used to calculate air flows and pressure losses in the circular duct system. The exhaust system is similar to the supply side and operated as a slave and therefore only the supply side is analyzed. A layout of the system is seen in figure 2.

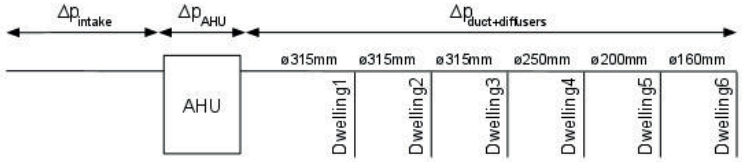

Figure 2: Layout of the supply side of the analyzed system including intake, AHU and duct diameters of the distribution system

System curves that show how system losses change with the air flow are made. The curves include all resistances along the flow path from intake to supply diffusers. The pressure loss between two points along the flow path can be expressed by eq. 2 (Schild et al. 2009):

$$\Delta p = k \cdot q^n \tag{2}$$

Where k is a constant that depends on system design, q is the air flow and n is the flow exponent that is 1 for fully developed laminar flow and 2 for fully developed turbulent flows. The pressure loss in the distribution system ($p_{duct+diffuser}$) was calculated for a representative selection of the 729 (3^6) possible combinations and locations of qL, qB and qF flows.

6.1 Intake and AHU

The flow exponent of the AHU is typically 1.4 (Schild et al. 2009) and we assumed a flow exponent of 2 for the inlet corresponding to typical manufacturer data. The constant k was calculated at 168l/s (corresponding to basic flow to all dwellings) through the components using resistances of $\Delta p_{AHU,168\ l/s}$=70 Pa and $\Delta p_{intake,168\ l/s}$=5 Pa. Figure 3 shows the pressure loss in the intake and AHU as a function of the flow.

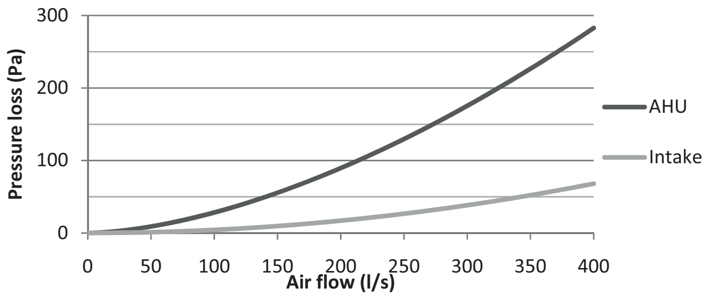

Figure 3: Pressure loss in AHU and intake as a function of the air flow

6.2 Diffusers

The two non-motorized diffusers in figure 4 were analyzed: A circular diffuser with a fixed outlet area and a diffuser with self adjusting vanes that change position as the pressure upstream of the terminal changes and thereby maintain a relatively stable inlet velocity. The terminal with variable outlet area is denoted an *active diffuser* the other a *circular diffuser*.

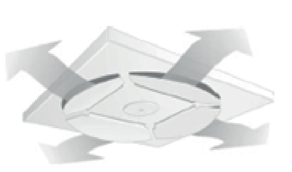

Figure 4: Investigated non-motorized diffuser. Left figure: Circular air diffuser with fixed opening area. Right figure: Active air diffuser with self adjusting inlet vanes that changes the outlet area depending on the pressure upstream of the diffuser

The flow exponent of the circular diffuser is 2 and the flow exponent of the active diffuser is approximately 1.3 according to manufacturer data (Acticon A/S 2011). The throw of the diffuser was estimated by literature where the throw of a radial jet that flows along a surface is calculated by eq. 3 (ASHRAE 2009).

$$x = \frac{K \cdot q}{v_x\sqrt{A_0}} \tag{3}$$

Where x is the throw, v_x is the throw velocity usually taken as 0.2 m/s, K is a constant depending on the design of the diffuser, A_0 is the outlet area and q is the airflow. The relative throw, x_r, of the diffuser with a constant outlet area assuming K remains constant is calculated by:

$$x_r = \frac{\mathrm{x}}{x_{max}} = \frac{q}{q_{max}} \tag{4}$$

The throw of a diffuser located in the middle of a room should not exceed half the room width plus the distance from the ceiling to the occupied zone (1.8 m above floor level) to avoid problems with draught. In a room that is 4 m wide and 2.5 m high the throw can be 2.7 m. The total air flow to the dwelling is split in two equal amounts to the living and bedroom and each diffuser supplies 3.5 l/s, 14 l/s and 28 l/s at low, basic and forced air flow respectively. We assume the throw of both diffusers are 2.7 m at 28 l/s. The relative throw of the circular diffuser is 0.5 at basic flow. This throw is shorter than half the room width and may cause draught. The active diffuser's throw varies less due to its variable outlet area and draught issues are expected to be less. The throw at low flow is not relevant regarding draught as residents are absent. The throw of both diffusers at basic flow and their effect on the indoor environment (e.g. noise, dumping of cold air at low supply temperature) should be investigated further to disclose if the performance of the diffusers are acceptable. Regarding energy consumption it was assumed that both diffusers require 20 Pa to supply 14 l/s. Figure 5 shows the total pressure needed immediately upstream of one circular and one active diffuser to supply a given air flow.

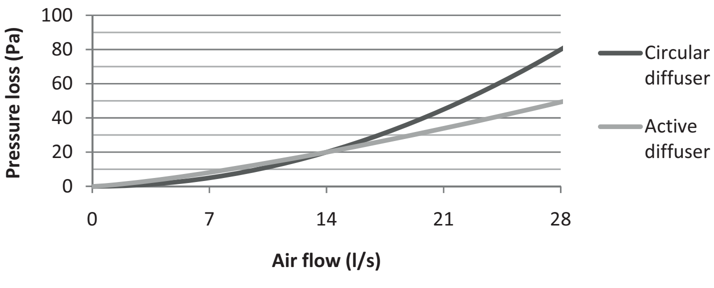

Figure 5: Pressure loss in one circular and one active diffuser as a function of the air flow

6.3 System with fixed static pressure control

In a system with fixed static pressure the location of the static pressure sensor impacts the power consumption of the fan. Figure 6 shows the system curve when the sensor was located between dwelling 5 and 6. The static pressure set point was fixed at 92 Pa in the system with the circular diffuser and at 61 Pa in the system with the active diffuser.

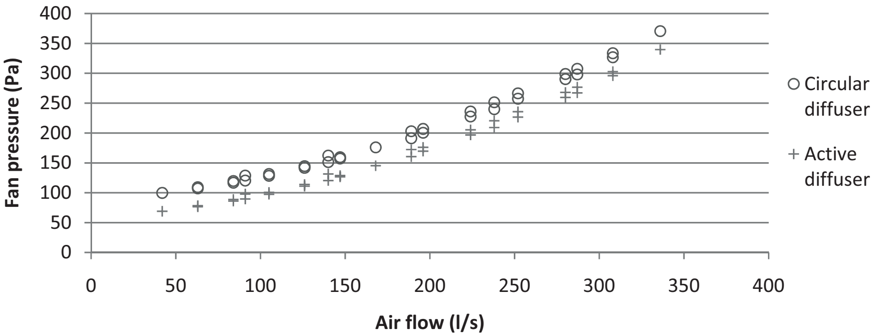

Figure 6: Fan pressure required to supply a given air flow in a system with fixed static pressure in the main duct between dwelling 5 and 6. The static pressure was fixed at 92Pa for the circular diffuser and 61Pa for the active diffuser.

6.4 System with ideal reset of the static pressure

In a system with ideal reset of the static pressure the minimum fan pressure needed to distribute the air depends on the air flows and their location in the distribution system. The same total flow can be obtained by different air flow combinations that result in different fan pressures. System curves including all losses from intake to supply diffusers are given in figure 7.

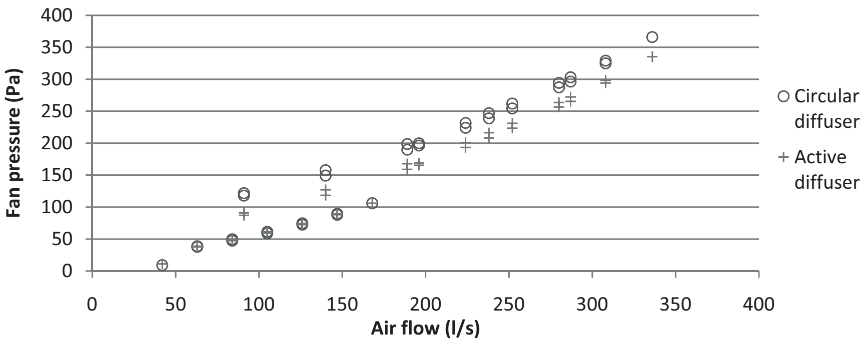

Figure 7: Fan pressure required to supply a given air flow in a system with ideal reset of the static pressure.

6.5 System with SPR schedule and preset damper positions

A pressure schedule is used in combination with predetermined damper positions to reset the static pressure set point at part load. A control algorithm collects flow requirements of each dwelling and uses the maximum requirement to set the pressure. The issue of detection of a dwelling flow requirement is beyond the scope of this paper, but it is assumed to be incorporated in the control algorithm. The pressure can switch between three levels (p_F, p_B and p_L) corresponding to the three possible air flows to a dwelling (q_F, q_B and q_L). These levels depend on both the air flow and on the location of the pressure sensor. In this example the sensor was located in the main duct between dwelling 5 and 6. The highest pressure level, p_F, is therefore identical with the pressure level in the system with fixed static pressure control that was set high enough to avoid that dwellings are starved of air flow. The dwellings with the highest flow requirement in each of the 729 possible air flow combinations have fully open dampers. Therefore one damper is always fully open and the pressure that must be throttled off somewhere in the system at part load is reduced. Dwellings that require air flows less than the maximum to one of the dwellings have dampers with partly closed positions. At least three positions must be predetermined for each damper when the flow to a dwelling can vary between three fixed air flows. The minimum number of positions is only possible when the air flow ratios are identical i.e. q_F/q_B equals q_B/q_L and the positions can be reused in the three pressure levels. Changing the static pressure scales the flow up or down but the ratio between the flows remains the same. The air flow ratios are not identical in the investigated system (q_F/q_B=2 and q_B/q_L=4), see table 2. An extra position must be introduced to provide the possible air flows at the three pressure levels. The positions are ranked in ascending order according to the flow ratios. Position 3 that only is used with pressure p_B is more open than position 4 but more closed than position 2.

Table 2: Static pressure set point, air flows, air flow ratios and damper positions in a system with preset damper positions

Static pressure set point (circular / active diffuser)	Air flows	Air flow ratios	Position
p_F (92Pa / 61Pa)	q_F	$\frac{q_F}{q_F} = 1$	1
	q_B	$\frac{q_F}{q_B} = \frac{56}{28} = 2$	2
	q_L	$\frac{q_F}{q_L} = \frac{56}{7} = 8$	4
p_B (23Pa / 23Pa)	q_B	$\frac{q_B}{q_B} = 1$	1
	q_L	$\frac{q_B}{q_L} = \frac{28}{7} = 4$	3
p_L (1.4Pa / 3.5Pa)	q_L	$\frac{q_L}{q_L} = 1$	1

Before determining the damper positions we need to consider the pressure variations occurring when the air flow to the dwellings change. Varying the flow to one dwelling affects the pressure in the entire system and thereby the flow to other dwellings. In systems with motorized control components these instabilities are adjusted for by the control component that continually adapts to the condition in the system. This is not possible in a system with predetermined damper positions. Pressure instabilities are instead managed by increasing the pressure drop in the branches making them less sensitive to pressure variations. The pressure drop, p_{ref}, needed in the branch to keep flow variations below ±10% is used in eq. 2 (k is given by: $\frac{p_{ref}}{q_{ref}{}^n}$) to express the pressure drop in the branch when the flow varies:

$$\Delta p_{+10\%} = \frac{\Delta p_{ref}}{q_{ref}{}^n} \cdot \left((1+0.1)\cdot q_{ref}\right)^n = \Delta p_{ref} \cdot (1+0.1)^n \tag{5}$$

$$\Delta p_{\div 10\%} = \frac{\Delta p_{ref}}{q_{ref}{}^n} \cdot \left((1-0.1)\cdot q_{ref}\right)^n = \Delta p_{ref} \cdot (1-0.1)^n \tag{6}$$

The difference between the two pressure drops calculated in eq. 5 and eq. 6 is used to express the pressure drop in the branches, p_{ref}, needed to keep flow variations below ±10% when the flow in the duct is predominantly turbulent i.e. n=2:

$$\Delta p_{ref} = \frac{\Delta p_{+10\%} - \Delta p_{\div 10\%}}{4 \cdot 0.1} \tag{7}$$

Flow variations are estimated to be ±10% if the pressure drop in the branches is 2.5 times higher than the pressure variations they experience. We apply this design criterion to our system design. The critical situations occur when the dampers have the lowest pressure drop i.e. when they are fully open. The maximum recorded pressure variation at a representative number of flow combinations is 8.5 Pa. This means that when dampers are fully open and have forced flow the branch should have a pressure drop of at least 21 Pa to avoid air flows more than ±10% of the desired flow. Both diffusers have pressure drops significantly above this value at maximum flow (see figure 5) and flow fluctuation due to pressure instability are not expected to be a problem.

The four positions of each damper are determined by applying the static pressure reset schedule to a system with free dampers. A representative number of flow combinations are analyzed and in each case the damper positions that balance the system perfectly are recorded. Even though large pressure variations are managed the calculated damper settings will vary slightly within each of the four positions. The average value of each damper's four positions is used further on in the calculations and will therefore cause both higher and lower air flows than expected. Dwelling 6 experiences least variation within one position because it is located close to the sensor.

The pre-calculated damper positions are applied to the system and air flows are calculated. All deviations were within ±10%. The system with the active diffuser has slightly higher flow deviations due to its lower pressure drop at maximum flow (see figure 5) but deviations are still within ±10%. System curves including all losses from intake to supply diffusers are given in figure 8.

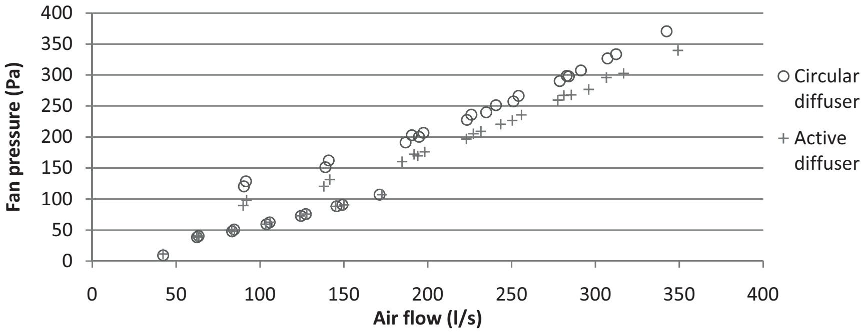

Figure 8: Fan pressure required to supply a given air flow in a system with SPR schedule and preset damper positions

7 Energy consumption

The annual energy consumption of the system was calculated from the generated system curves and a diurnal occupancy schedule. The schedule is adapted from standard 15665 (CEN 2009) and also based on typical estimated occupancies. For simplicity the same occupancy schedule is used all days of the year

Table 3: Diurnal occupancy and activity schedule for the 6 dwellings. 'UO' represent Unoccupied hours with low air flow (q_L), 'O' represent Occupied hours with basic air flow (q_B) and 'OA' represent Occupied hours with Activity and forced air flow (q_F).

Hour	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Dw1	O	O	O	O	O	O	OA	U	U	U	U	OA	U	U	U	U	U	O	OA	O	O	O	O	O
Dw2	O	O	O	O	O	O	OA	U	U	U	U	OA	U	U	U	U	U	O	O	O	O	O	O	O
Dw3	O	O	O	O	O	O	O	OA	O	O	O	OA	U	U	U	U	O	O	O	O	O	O	O	O
Dw4	O	O	O	O	O	O	OA	O	O	O	O	OA	O	O	O	O	O	O	OA	O	O	O	O	O
Dw5	O	O	O	O	O	O	OA	U	U	U	U	U	U	U	U	O	O	OA	O	O	O	O	O	O
Dw6	O	O	O	O	O	O	O	OA	U	U	U	U	U	U	U	U	O	O	O	O	OA	O	O	O

The energy consumption for transportation of the air is expressed by the power consumed by the fan:

$$P = \frac{q\Delta p}{\eta_{total}} \tag{8}$$

The total efficiency of the fan at a fixed speed was deduced from manufacturer data. As the fan speed is altered the mechanical efficiency decreases and the total efficiency should be reevaluated. Typical efficiency data for an EC motor was used (Schild et al. 2009) to recalculate the total efficiency at reduced speed. The total efficiency at speeds corresponding to low, basic or forced flow in all dwellings is given in figure 9.

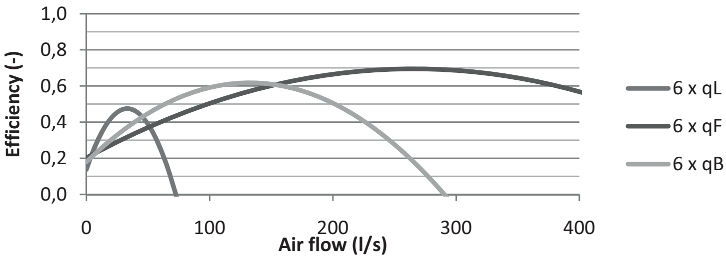

Figure 9: Total efficiency at the three speeds corresponding to forced flow in all dwellings (6 x q_F), basic flow in all dwellings (6 x q_B) and low flow in all dwellings (6 x q_L)

The annual energy used to transport the air is seen in figure 10 assuming supply and exhaust system are identical. The systems SFP values at maximum flow are given on the bars.

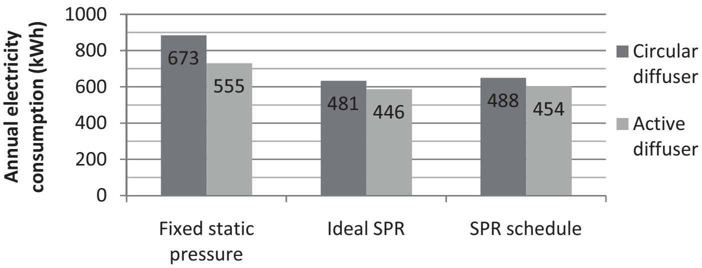

Figure 10: Annual electricity consumption for ventilation of the 6 dwellings. The systems SFP at maximum flow are given on the bars

8 Discussion

The advantage of the SPR strategy is that it saves energy at part load conditions compared to a system with fixed static pressure control. Of the analyzed cases the control strategy with fixed static pressure sets the upper boundary for the ventilation system's annual electricity consumption and the system with ideal reset of the static pressure sets the lower boundary. The system with the reset schedule and preset damper positions has slightly higher electricity consumption than the ideal SPR strategy. The electricity consumption is approximately reduced by 30% in the systems with circular diffusers and 20% in the systems with active diffusers when resetting the pressure at part load conditions compared to systems with the same diffusers and a fixed static pressure control. The savings are in the lower range of the saving reported in literature of 30% to 50%.

The system with preset damper positions requires accurate setting of the static pressure to deliver the desired flows. Even small deviations from the required pressure can result in significant flow changes. The lowest pressure level is only used when all apartments are unoccupied and the situation will probably seldom occur. The level will most likely be unattainable to hold and a tradeoff can be to design the control algorithm to maintain a higher pressure and not have fully open dampers at this air flow situation.

Balancing and commissioning of the system is time-consuming because the predetermined damper positions needs to be tuned. It is often advised to minimize the need for manual balancing of the system (Svensson 2008) by using automatically adjusting control components. However, tuning of systems with automatic control components is also needed to obtain optimal performance. Rational procedures to balance/tune a system with preset damper positions should be investigated in future work. Implementation of the SPR schedule with preset damper positions requires dampers that easily can switch between the four predetermined positions after receiving a signal from the control algorithm. A damper fulfilling this criterion should be developed. If the cost of automatic control components decreases the reset schedule could use these to switch between the three flows.

The two investigated diffusers result in very small difference in electricity consumption when the SPR strategy is employed. The difference is larger with fixed static pressure control. The preferred diffusers should be the one that provide best energy and indoor environmental performance. The diffusers throw at basic flow and the impact on the indoor environment should be investigated in future work to determine the best performance. The active diffuser that changes the outlet area depending on the upstream pressure is an interesting component for variable air flow ventilation in residences.

9 Conclusion

This paper investigated a simple and inexpensive solution for system design of a centralized DCV system in multi-family dwellings. An economical estimate showed that the initial cost of implementing DCV in a system with an efficient heat exchanger should not exceed 3400 DKK per dwelling. The system design focused on the control of the system. Resetting the static pressure at part load conditions reduced the annual electricity consumption by 20% to 30% compared to a control strategy with fixed static pressure.

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Most standards and buildings codes specify desired levels of indoor air quality through constant ventilation rate requirements. The Danish Building Code requires a constant air flow rate equivalent to at least 0.5 air changes per hour in residential buildings. Constant air flow requirements are inconsistent with the time varying needs for ventilation of residential buildings that depend on occupancy, pollutant emission etc. and results in periods with poor air quality and/or unnecessary energy consumption. If the ventilation rate is varied according to the demand, the indoor climate can be improved and the energy consumption for ventilation can be reduced compared to a system with constant air flow.

In this thesis, aspects of demand specifications and system design of demand controlled ventilation for residential buildings were studied. The results of the project provide more flexible approaches to ventilation design for residences that allow occupancy based DCV approaches to comply with codes and standards that are currently based on continuous ventilation rates. Furthermore, a simple, cost-effective and energy-efficient system design for DCV in multi-family dwellings is propose.

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